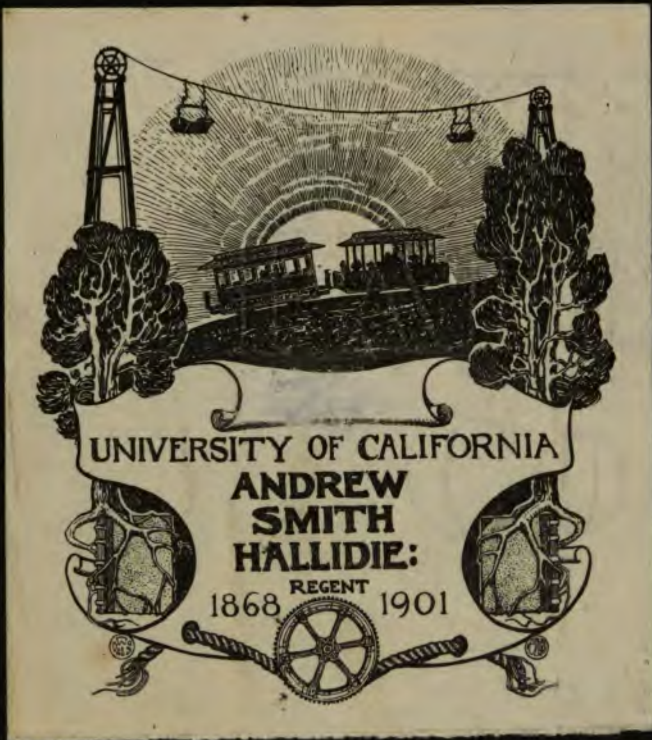
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THE STEAM TURBINE

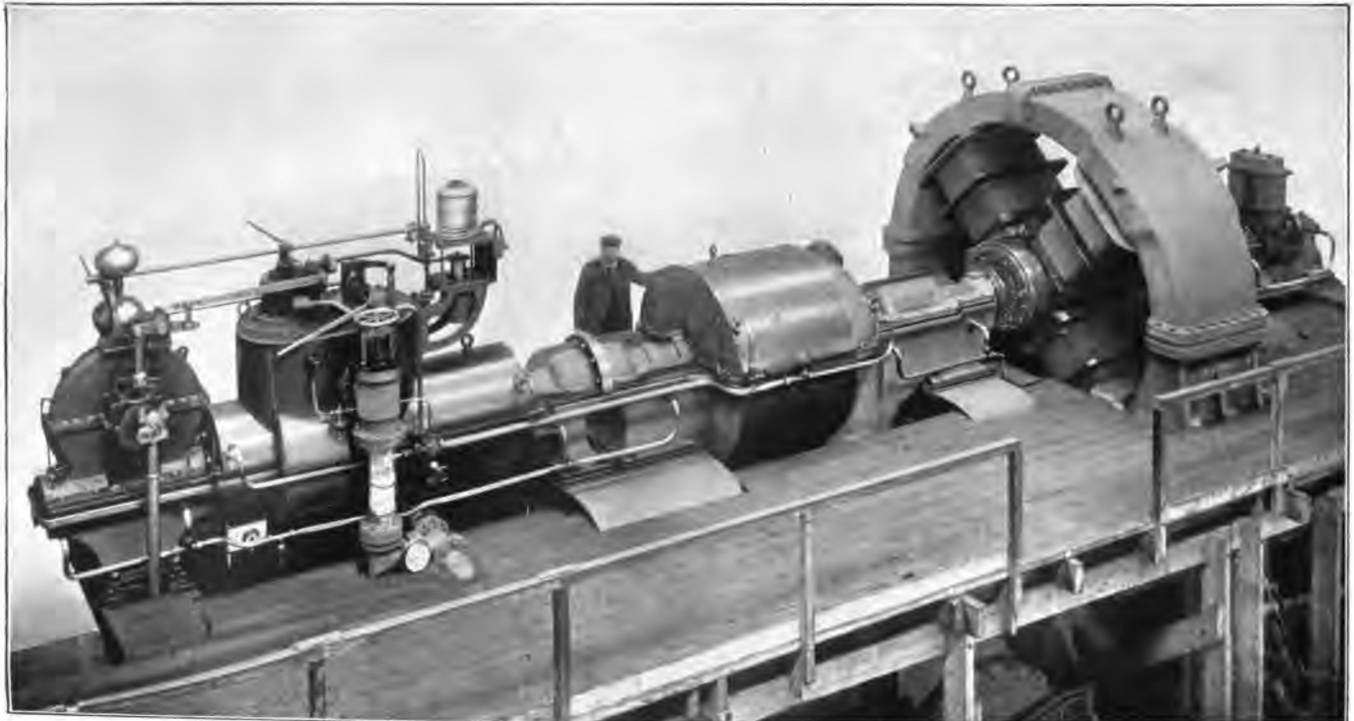


PLATE I.—1000-KILOWATT PARSONS TURBO-ALTERNATOR FOR ELBERFELD CORPORATION, AS ERECTED FOR THE TRIALS AT THE HEATON WORKS, NEWCASTLE-ON-TYNE.

THE

STEAM TURBINE

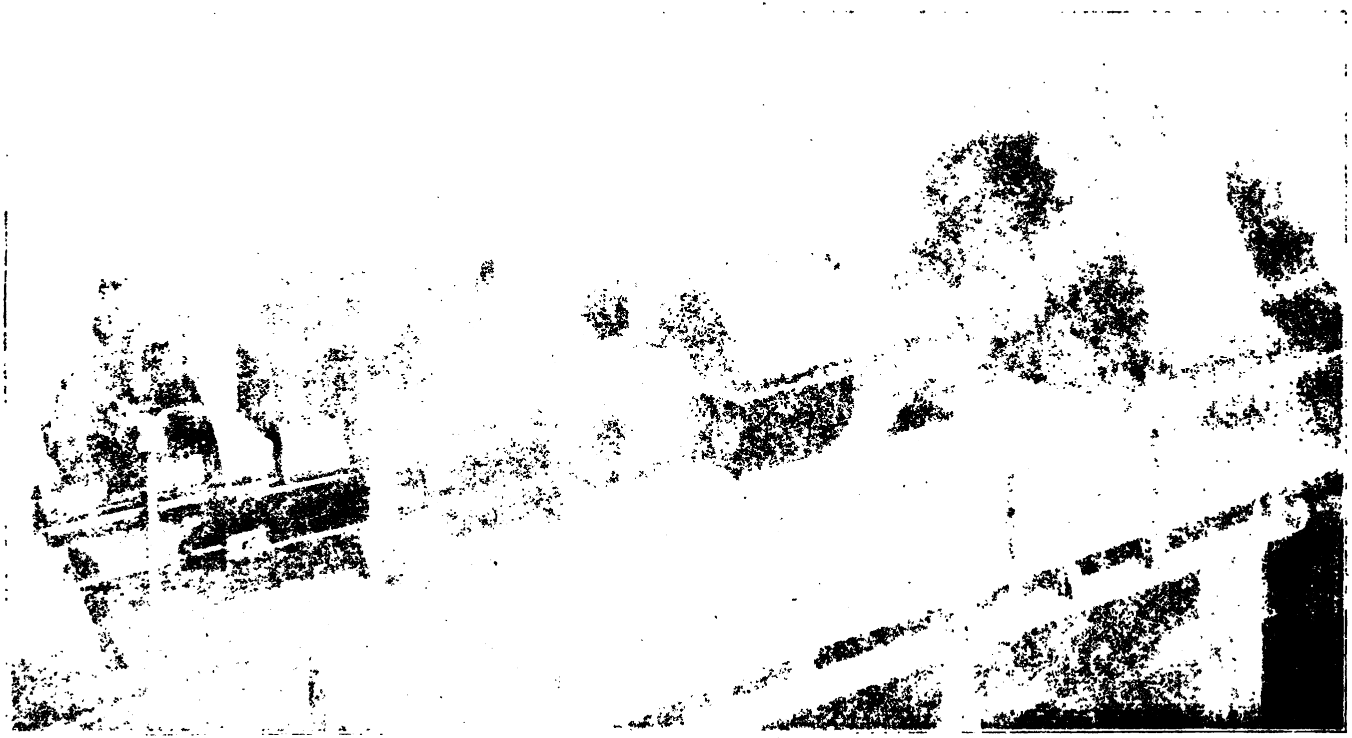
BY

ROBERT M. NEILSON

WH[illegible] EXHIBITION[illegible] ASSOCIATE MEMBER OF THE [illegible]
[illegible] MEMBER OF THE MANCHESTER [illegible]
[illegible]OW OF THE [illegible]ERED INSTITUTE OF [illegible]
LECTURE[illegible] STEAM AND THE [illegible] ENGINE AT THE [illegible]
SCH[illegible]

WITH NUM[illegible] ILLUSTRATIONS

LONGMANS, GREEN, AND CO.
39 PATERNOSTER ROW, LONDON
NEW YORK AND BOMBAY
1902



... 1.—1000-Kilowatt Pa... ...Corporation, as erected for the Trials at the ...Works, Newcastle-on-Tyne.

THE STEAM TURBINE

BY

ROBERT M. NEILSON

WHITWORTH EXHIBITIONER; ASSOCIATE MEMBER OF THE INSTITUTION OF MECHANICAL ENGINEERS; MEMBER OF THE MANCHESTER ASSOCIATION OF ENGINEERS; FELLOW OF THE CHARTERED INSTITUTE OF PATENT AGENTS; LECTURER ON STEAM AND THE STEAM ENGINE AT THE HEGINBOTTOM TECHNICAL SCHOOL, ASHTON-UNDER-LYNE

WITH NUMEROUS ILLUSTRATIONS

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PREFACE

THAT the steam turbine is likely to be extensively used in the future is admitted by most engineers; but, although a good deal has lately been written about this type of engine, this literature has mostly consisted of descriptions of the principal features only, or of accounts of the results of tests.

The author has endeavoured in this book to describe, not only the principal parts of the leading types of steam turbine, but also the small details which, in the case of this motor, have such a preponderating influence in determining success or failure. The theory of the action of the steam turbine is also treated of, and the subject is likewise dealt with historically.

Comparisons have necessarily been made with the hydraulic turbine and with the reciprocating engine; but, with a view to extending the usefulness of the book, the author has assumed on the part of the reader no prior knowledge of the hydraulic turbine, and only an elementary knowledge of the reciprocating engine and of the laws of thermo-dynamics.

With a like object in view the author has tried to make the mathematical reasoning as simple as possible.

As entropy-temperature diagrams are not yet widely understood, a chapter on this subject has been given; but the matter has been treated as briefly as possible.

The results of tests of steam turbines given throughout the book have been carefully selected with a view to obtaining the strictest accuracy.

The author takes this opportunity of thanking the various individuals and firms who have given him information and assistance, and of expressing his indebtedness to Messrs. C. A. Parsons and Co., Newcastle-on-Tyne, and the Société de Laval of France for the loan of several blocks.

R. M. N.

30, Cross Street, Manchester,
June, 1902.

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THE STEAM TURBINE

CHAPTER I.

GENERAL REMARKS ON TURBINES.

A TURBINE is a machine in which a rotary motion is obtained by the gradual change of momentum of a fluid.

Fig. 1 shows a turbine diagrammatically. The partitions B between the passages A are called vanes, or blades, or buckets. Now, it is obvious that, if a fluid enters the space between two vanes in the direction shown by the arrow 1, and leaves in the direction shown by the arrow 2, the component of its velocity perpendicular to the radius will gradually change in its passage. The component might not change during the whole of the passage of

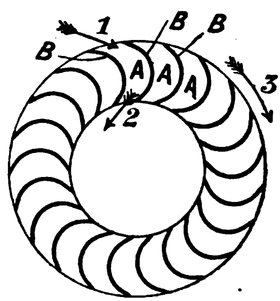


FIG. 1.—Diagrammatic Illustration of Turbine.

the fluid owing to the vanes themselves having a velocity; but it will have a gradual change during at least some part of this passage. The fluid, therefore, has its momentum gradually changed, and it is this change of momentum which causes the vanes to rotate. The turbine wheel in the figure would rotate in the direction of the arrow 3. The action of the fluid on the

turbine will be discussed more fully later on; it is only desired at present to give a general idea of a turbine.

Turbines may be classified in several ways. Firstly, they may be classified according to the actuating fluid. The fluids most commonly used are water and steam, and the turbines actuated thereby are called respectively hydraulic turbines and steam turbines.

Turbines may be classified according to the direction of flow of the fluid into three classes: (1) In **radial-flow turbines** the fluid travels from the centre to the circumference of the wheel, or from the circumference to the centre. This class is subdivided into **outward-flow** and **inward-flow** turbines, according as the fluid passes from the centre to the circumference, or from the circumference towards the centre. (2) In **parallel-flow** or **axial-flow turbines** the direction of the flow of the fluid is parallel to the axis of the wheel, or in a spiral co-axial with the wheel. (3) In **mixed-flow turbines** the fluid flows both as in a radial-flow and as in a parallel-flow turbine.

Turbines are classified in other ways besides these; but as the other ways are not of importance, or do not hold good with steam turbines, we shall not refer to them.

Fig. 2 illustrates the principle of a **parallel-flow De Laval** steam turbine. The steam reaches the wheel by way of the divergent nozzles, where it expands and attains a great velocity. With this velocity it impinges on the vanes of the wheel, and causes the latter to rotate at a high speed. The wheel is enclosed loosely in a box or case, from which the steam escapes to the atmosphere or to a condenser. A section of one of the nozzles is shown at Fig. 2A drawn to an enlarged scale. In this figure the dotted line indicates the axis of rotation of the

wheel. The De Laval turbine will be more fully described later on.

FIG. 2.—Action of Steam in De Laval Turbine.

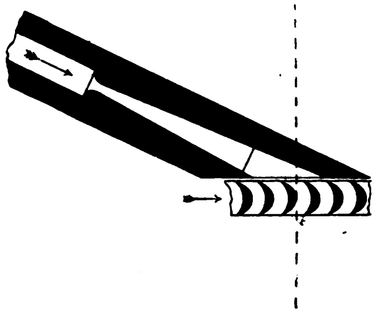


FIG. 2A.—Section of Nozzle of De Laval Steam Turbine.

Figs. 3, 4, 5, 6, 7, and 8 illustrate parts of a **Parsons parallel-flow** steam turbine. In this turbine the steam acts successively on a number of rings of blades. Part of one of these is shown in perspective view in Fig. 3, in elevation in

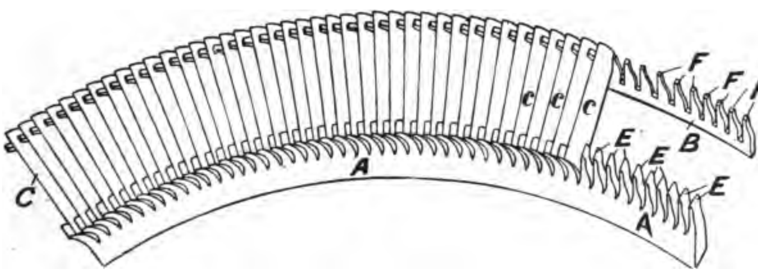


FIG. 3.—Blades and Shrouds of Parsons Parallel-flow Steam Turbine.

Fig. 4, and in plan in Fig. 5. Each ring of blades in this example is formed of blades, *c*, gripped in suitable recesses in shrouds, A and B. The rings thus formed are fixed alternately to the inside of the fixed cylindrical casing of the turbine, and to a revolving drum mounted inside the casing. Figs. 6

and 7 show parts of the casing and drum, the casing being lettered I and the drum H. Fig. 6 is a section taken through

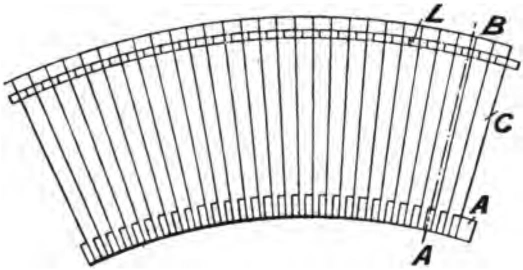
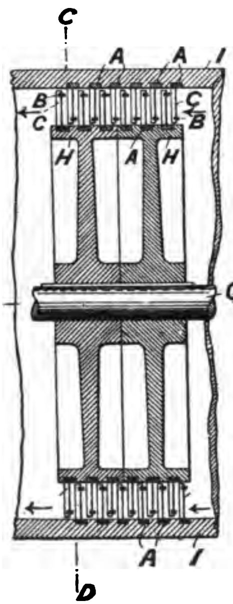


FIG. 4.

FIG. 5.

Blades and Shrouds of a Parsons Parallel-flow Steam Turbine.

the axis of the casing, while Fig. 7 is a cross-section on the line CD of Fig. 6. Power is obtained from the spindle G,



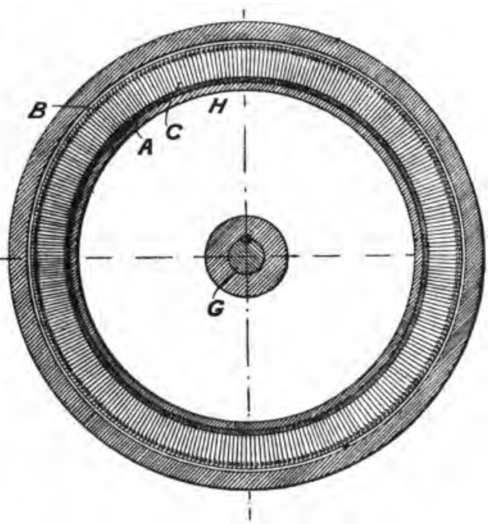


FIG. 6.—Partial Axial Section. FIG. 7.—Cross-section.
Parsons Parallel-flow Steam Turbine.

on which the drum H is keyed. It will be seen that the larger shroud A of each ring is secured to the casing or drum, while the smaller shroud B is free. The steam passing in the direction of the arrows in Fig. 6 acts on the moving blades so as to rotate them, and with them the drum and spindle. The fixed blades serve as guides to cause the steam

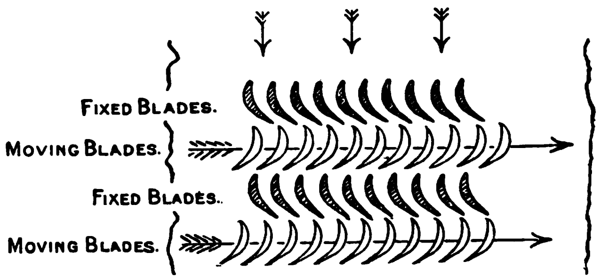


FIG. 8.—Action of Steam on the Blades of a Parsons Turbine.

after leaving one ring of moving blades to impinge in the right direction on the next ring of moving blades. The action of the steam on the blades can be clearly seen in Fig. 8, where the horizontal arrows show the direction of motion of the moving blades and the vertical arrows the direction of flow of the steam. It should be pointed out that the clearance between the fixed and the moving blades is very small—not nearly so great as is shown for the sake of clearness on the drawings.

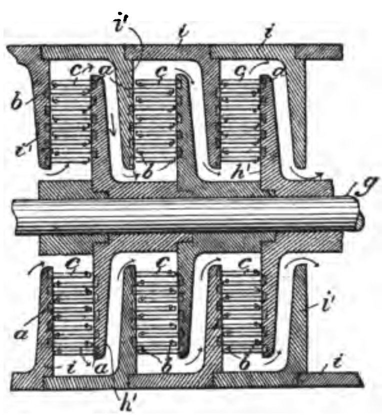


FIG. 9.—Parsons Radial-flow Steam Turbine, Partial Axial Section.

Fig. 9 is a partial axial section through a **Parsons radial-flow**

turbine, and Fig. 10 illustrates a ring of blades for the same drawn to an enlarged scale. The blades *c*, both fixed and moving, are held in shrouds, *a* and *b*, of a similar nature to the shrouds A and B of the parallel-flow turbine. The cylindrical casing *i* carries internal annular flanges, *i'*, to which are attached the larger shrouds *a* of the fixed rings of blades; while the similar shrouds of the moving rings of blades are supported on annular flanges, *h'*, carried by the spindle *g*. The smaller shrouds *b* of

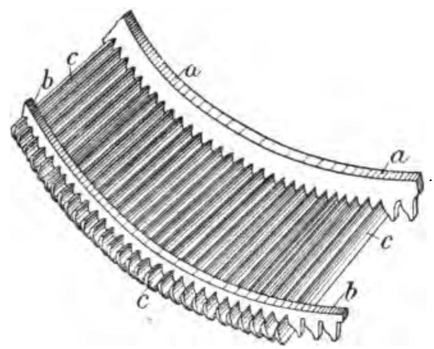


Fig. 10.—Blades and Shrouds of a Parsons Radial-flow Steam Turbine.

both fixed and moving rings are left free. The path of the steam is indicated by the arrows in Fig. 9, and it will be seen that the steam acts on the moving blades while flowing radially outwards in several stages.

The Parsons turbine in its several forms will be more fully described afterwards. The short description just made will, however, give a general idea of its nature.

CHAPTER II.

HISTORY OF THE STEAM TURBINE.

GOING back long before the days of Watt and Newcomen, we find a reaction steam-engine mentioned by the Egyptian philosopher **Hero** in his book on "Pneumatics," written in the second century B.C. This engine consisted of a hollow sphere rotating on two trunnions, through one of which it received steam from a generator situated below the sphere. The sphere was provided with two opposite projecting arms at right angles to the axis of the trunnions, the arms being furnished each with a nozzle at right angles to the arms and to the plane containing the arms and the trunnions. The nozzles were pointed in opposite directions, and the steam which escaped by them from the sphere caused the rotation of the latter about the trunnions.

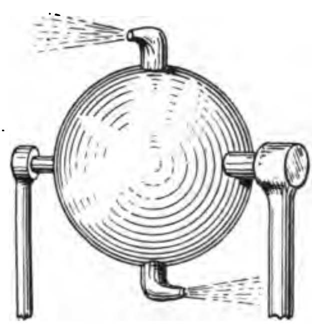


FIG. 11.—Hero's Rotating Steam Globe.

In A.D. 1577 a German mechanic is said to have used Hero's engine to rotate a broach in place of a turnspit.

In 1629 an Italian architect named **Branca** described a steam wheel or turbine in which a jet of steam was projected against a series of vanes on a rotating wheel.

In 1642 a Jesuit named **Kircher** used Branca's wheel, but with two jets of vapour acting on its circumference instead of only one.

In 1784 **Wolfgang de Kempelen** was granted a British patent for "Obtaining and transmitting motive power." The patentee thus describes his invention—

"When the machine acts by boiling water, or rather the vapour proceeding therefrom, a boiler is to be constructed (A, Fig. 12) furnished with a valve of security (B), the weight

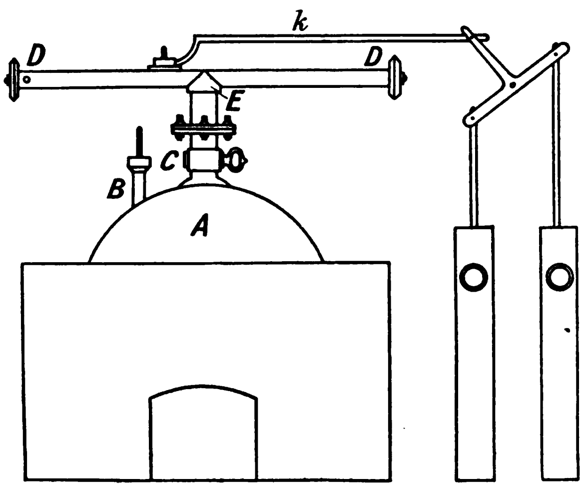


FIG. 12.—Wolfgang de Kempelen's Turbine.

of which is to be proportioned to the strength of the machine. At the upper extremity of the boiler is to be fixed a turn-cock (C), upon which the cylinder (DD) is to be screwed, the form of which cylinder appears in Fig. 13, where DD is a hollow cyclinder or tube, in the centre of which E is an aperture to contain the worm of the screw. FF is a tube of cast iron, having at the lower extremity a circular projection or plate, which, when this tube is pushed into the other

tube, GG, fills up the cavity therein marked (*aa*), so that the screw (*bb*) extends beyond the utmost length of the tube GG. Upon this screw the cylinder DD, with its nut, is to be fixed, and upon the plate of the tube GG of brass is to be screwed another plate (HH) of equal dimensions, so that the little plate, when it is in the cavity (*aa*), may be enclosed between

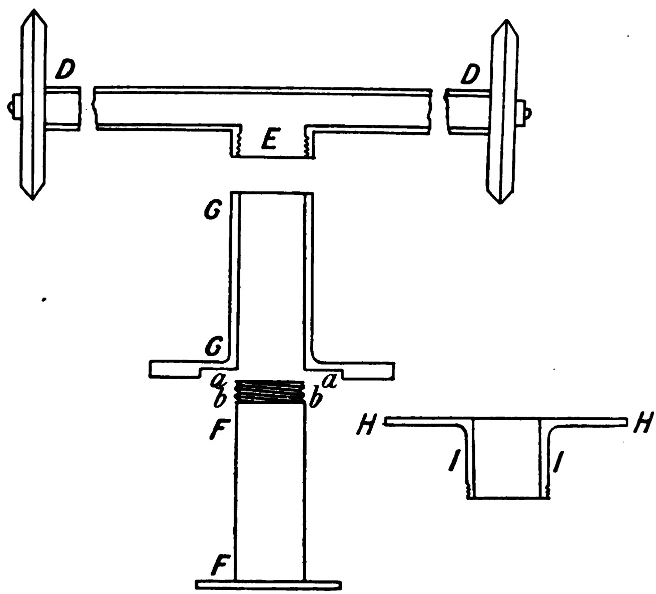


Fig. 13.—Details of Kempelen's Turbine.

two plates, and the tube FF left at liberty to turn round. The plate HH has also a short tube (II) of an equal aperture with the tube GG, and at the end of it a screw is fixed, which surrounds the cock C, and is fastened within. Near each extremity of the cylinder DD, but on the contrary sides, is a small aperture, the size of which must be commensurate to the extent of the superfices of the boiling water, as, for instance, when the boiler measures within six feet in diameter,

requiring a valve of security weighing five pounds, the aperture near each end of the moving cylinder must be one inch in diameter. To put the machine in motion when the vapour of the boiling water is found strong enough to lift up the valve, the cock (C) is to be opened; the vapour instantly rushes through, and fills the cylinder DD, and finding a vent through the small apertures near its extremities on different sides, drives the cylinder round by reaction with exceeding great velocity. Having accomplished this first moving power which constitutes the principle of the machine, any kind of machine or engine may very easily be put into motion by it by means of a handle crown-wheel pinion, or other connection adapted to it, as is done with respect to a double pump by the excentric trunnion, *k*, Fig. 12."

The patentee then describes in his specification how his engine can be worked by water conveyed from a height, or by water acted on by steam pressure. The last-mentioned method is not illustrated, but the patentee states that two receivers of iron or copper must be provided between the boiler and the turning cylinder, and connected with both. The steam from the boiler is admitted alternately to the two receivers, and, pressing on the surface of the water, forces this into the turning cylinder, and rotates the latter by its reactive force when issuing from the apertures at its ends. The water is returned to the receivers.

In the same year **Watt** was granted letters patent for certain improvements relating to steam-engines. Most of the improvements relate to reciprocating engines, but one improvement relates to a rotary engine or turbine. This engine, or turbine, is described and illustrated in one of its "most commodious" forms by Watt in his specification. A vessel, ABDEC,

is rotatable on a pivot resting on the support J (Fig. 14), and is also supported by a collar, K, at its upper end. The vessel has a vertical partition, which divides it into two chambers, and each chamber has an aperture, R, at its upper end, which can communicate with a pipe, L (Figs. 14 and 15), conveying steam from a boiler. The rotating vessel is enclosed in a containing tank or vessel, MN, which is nearly filled with mercury, water, oil, or other liquid; and valves, F, G, are provided to allow

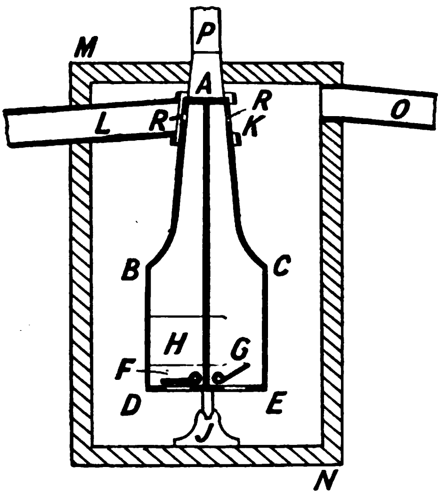


Fig. 14.

Fig. 15.

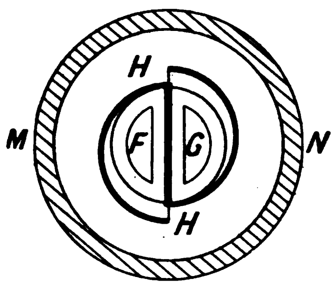


Fig. 16.

Watt's Turbine.

this liquid to enter the two chambers of the rotating vessel. Fig. 16 is a sectional plan of the rotating vessel and the enclosing tank. Openings, H (Figs. 14 and 16), are provided in the sides of the rotating vessel near the bottom.

Steam enters one of the chambers of the rotating vessel through its aperture R, and forces the liquid out of the chamber into the tank by way of the hole H, the valve F or G, as the case may be, being kept closed by the pressure of the steam. The reactive force of the jet issuing from H

rotates the vessel. While the steam is entering one chamber of the rotating vessel, the steam from the other chamber is exhausting by its aperture R into the atmosphere, or into the tank to be conveyed by the pipe O to a condenser. The escape of the steam from either chamber allows the liquid in the tank to enter that chamber by the foot-valve F or G. Power for driving machinery is got from axle P. In Watt's specification drawing the rotating vessel is shown as being about 12 inches in diameter by about 30 inches high, measured to the top of the steam-pipe.

It will be seen that this turbine is the same in principle as the last-mentioned form of De Kempelen's turbine, but as Watt's specification was signed and sealed by him only about a month after De Kempelen's, and as he had been granted his patent a few months previously, it seems probable that he devised his turbine quite independently of De Kempelen.

Since the days of James Watt, a great number of patents have been granted for inventions relating to steam turbines. A selection has been made of those which the author considers most interesting and most important, but only a very small proportion of those of recent years can of course be noticed.

In 1791 **James Sadler**, an engineer of the city of Oxford, was granted a patent for an invention entitled, "An engine for lessening the Consumption of steam and fuel, in steam or fire engines, and gaining a considerable Effect in Time and Force." The drawings enrolled with the specification are here reproduced, and the inventor's "Explanation" is also given in full. The latter is as follows: "Fig. 1st (Fig. 17). The Steam generated in the Boiler A is convey'd by y^e Steam pipe B into y^e spindle of y^e rotative Cylinder C which is left

hollow for that purpose & connected with ye pipe B by means of a stuffing Box at N which admits of the rotative motion of ye spindle without loss of Steam, it thère passes along ye Arms of ye rotative Cylinder nearly to ye ends thereof where it meets with a jet of cold Water whereby it

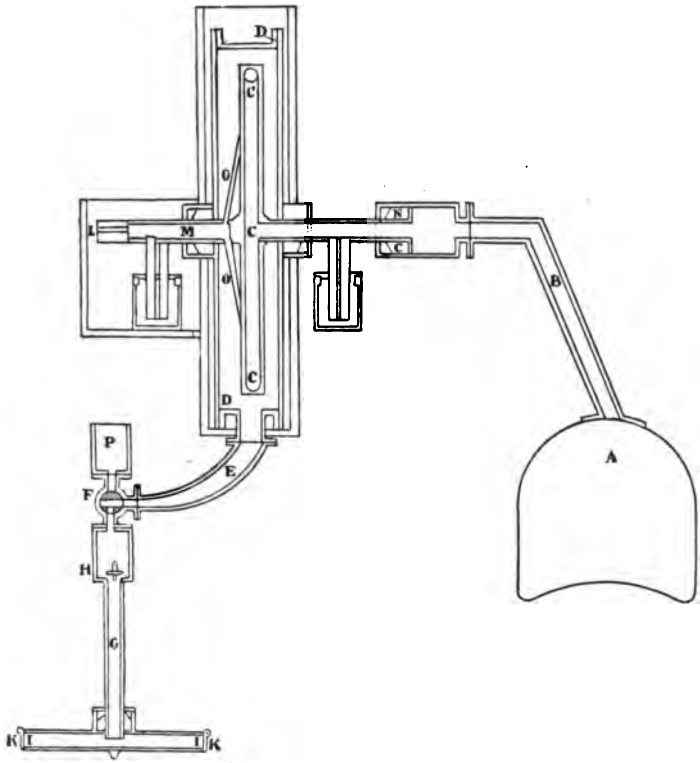


Fig. 17.—Sadler's Engine.

is condensed this jet is introduced by ye small pipes OO which communicates with ye spindle M which is hollow and receives ye Water by a hole at L, the Water falls thro' ye bottom of ye case DD into ye pipe E and is together with ye air admitted into ye pipe G thro' ye Cock F and descending when ye valve H is open into ye pipe I which has a

rotative motion round y^{e} end of y^{e} pipe G, it is thereby ejected thro' y^{e} valves KK the air which is left in y^{e} upper end of y^{e} pipe G is by turning y^{e} cock F suffer'd to escape whilst an equal portion of Water takes its place out of the Reservoir P, Otherways y^{e} steam is admitted into y^{e} Case DD, and rushing into the Arms of y^{e} rotative Cylinder is therein Condensed whilst y^{e} external steam by its action on y^{e} Arm causes a rotative motion—these Arms may also be included in y^{e} Boiler A which will prevent the necessity of a Case. Fig. 2nd (Fig. 18) Is a Section of y^{e} Machine across y^{e} spindle of y^{e} rotative Cylinder before described & AA are two small pipes which convey the Cold water for injection into y^{e} ends of y^{e} Cylinder Arms at BB.

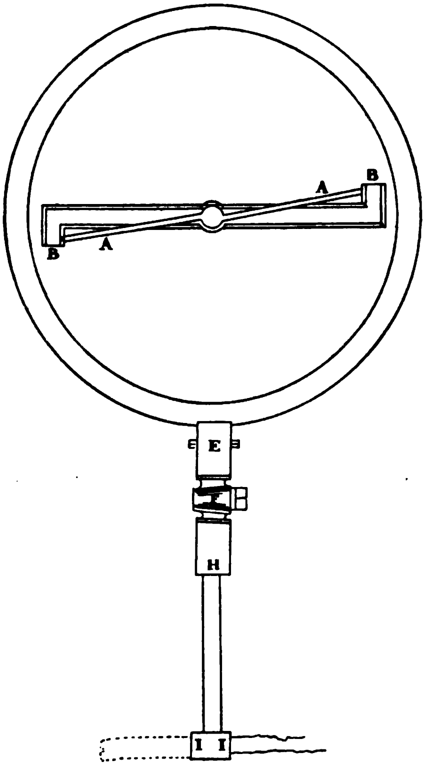


FIG. 18.—Cross-section of Sadler's Engine.

which as described before passes down y^{e} pipe E thro' y^{e} Cock F and valve H into y^{e} rotative arms II it is ejected from them by y^{e} valves KK as before described."

Noble's Patent, No. 3289 of 1809. A drawing from the specification relating to this patent is here reproduced (Fig. 19). The accompanying description is not very good, but it

is gathered that steam proceeding from the boiler A by the pipe B impinges on the "catches and ratchets" of the wheel C, and forces the wheel to rotate in the direction of the arrow. The ratchet wheel E and pawl F prevent the possibility of a contrary rotation.

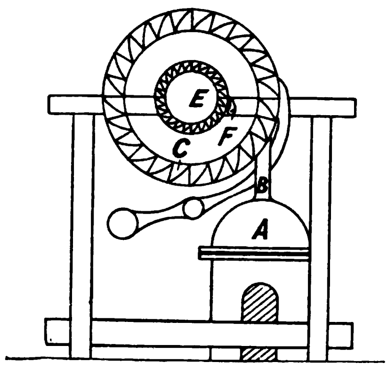


FIG. 19.—Noble's Steam Wheel.

Trevithick's Patent, No. 3922 of 1815. One part of this invention consists in "causing steam of a high temperature to spout out against the atmosphere, and by its recoiling force to produce motion in a direction contrary to the issuing steam similar to the motion produced in a rocket or to the recoil of a gun." The patentee, who seems fond of firearms as similes, states that the mode of carrying this part of his invention into effect will be readily understood "by supposing a gun-barrel to be bent at about a quarter of its length from the muzzle, so that the axes of the two limbs shall be at right angles to each other, and the axis of the touch-hole at right angles to the axis of the short limb, or the limb containing the muzzle. . . . Then in the top of a boiler suitable to the raising [of] steam of a high temperature, make a hole and insert the muzzle of the gun-barrel into that hole, so that the gun-barrel may revolve in the hole steam-tight, and let the short bend of the gun-barrel be supported in a vertical position by a collar which will permit the breech of the gun-barrel to describe a horizontal circle, the touch-hole being at the side of the barrel. If steam of a high pressure be then

raised in the boiler, it will evidently pass through the gun-barrel and spout out from the touch-hole against the atmosphere with a force greater or less according to the strength of the steam, and as the steam is also exerting a contrary force against that part of the breech which is opposite to the touch-hole, the barrel will recoil, and because the other end is confined to a centre the breech end will go round in a circle with a speed proportionate to the pressure given, and may be readily made to communicate motion to machinery in general." The patentee gives this explanation "merely to convey to the mind a clear idea" of his invention. In practice, he says, he uses more than one revolving arm, and he makes the aperture through which the steam is projected capable of being increased or decreased by means of a sliding piece worked by a screw. Several other variations may also, he states, be adopted.

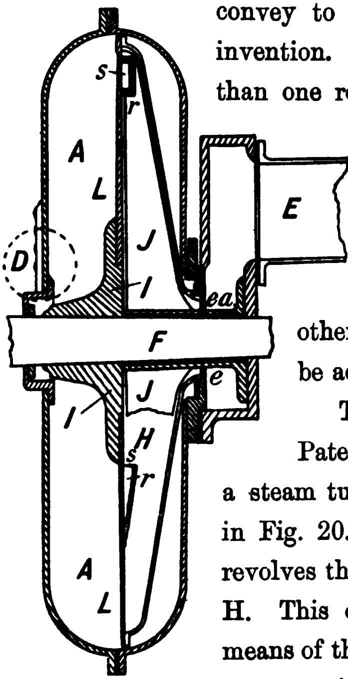


FIG. 20.—Ericsson's Turbine.

The specification of **Ericsson's** Patent, No. 5961 of 1830, describes a steam turbine, a section of which is given in Fig. 20. A is a fixed casing in which revolves the shaft F carrying the "fly-drum" H. This drum is attached to the shaft by means of the boss I and the plate L. Channels *r* are provided in the plate L, which channels open at *s* into the fly-drum. Vanes J are situated inside the fly-drum, but are not connected to it, being attached only to the fixed collar *a*. One of the channels

r is shown separately in Fig. 21. The channels are also shown in Fig. 22, which is a view at right angles to Fig. 20, and exhibits also the fixed vanes J. In Fig. 22, however, besides the channels r in the face of the fly-drum, channels r' are also shown in the periphery of the same. The steam enters the casing by the pipe D, and its action in passing into the fly-drum through the channels r causes the drum to rotate, while the fixed vanes J prevent the rotation of the steam which leaves the casing at e and passes away by the exit pipe E. The inventor states, towards the end of his specification, that the object of his invention would be equally well obtained if the steam were to travel in a reverse manner —that is, to enter the fly-drum at e and leave it by the channels r.

Fig. 21.

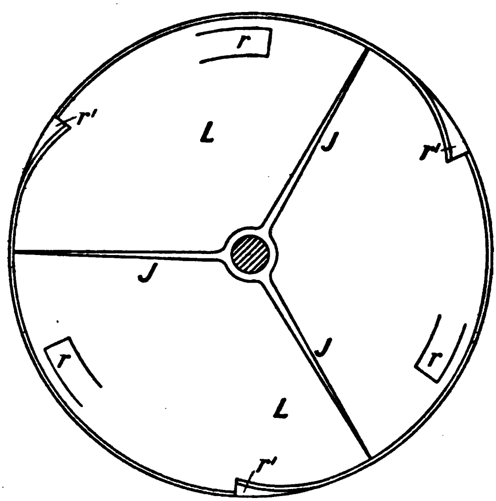


Fig. 22.—Vanes and Channels of Ericsson's Turbine.

Perkins' Patent, No. 7242 of 1836. The patentee states that in previous rotary steam-engines of the kind in which motion has been obtained by the reaction of steam-jets issuing from a rotating apparatus, the steam has been allowed to freely escape from the orifices into the atmosphere or into a steam chamber. In the patentee's engine, however, a series of

abutments, like the teeth of a ratchet wheel, are arranged in a ring for the steam-jets to impinge on.

The specification of **Pilbrow's** Patent, No. 9658 of 1843, is very interesting. The inventor seems to have experimented and theorized on the expansion and impulsive force of steam to a considerable extent. He found out, among other things, that, with a nozzle having an orifice three-eighths of an inch in diameter (the form of the nozzle is unfortunately not stated), the impulsive force of the steam issuing into the atmosphere was nearly proportional to the gauge pressure forcing the steam out. The pressures experimented with varied from 10 to 60 lbs. above atmosphere, and the impulsive force was measured "at the best distance from the orifice of the nozzle (about three-quarters of an inch)." With a gauge pressure of 60 lbs., the experimenter found that the total impulsive force (not the impulsive force per square inch) was about 14 lbs. Pilbrow calculated from this that the best velocity for the vanes of his turbine, using steam at 60 lbs. above atmosphere, would be about 1250 feet per second. He admitted that this was a very high velocity, but hoped to be able to utilize it.

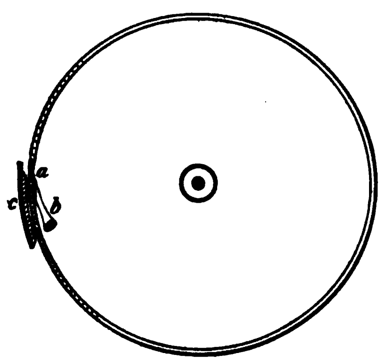


FIG. 23.—Simple Turbine of Pilbrow's.

Fig. 23 shows a simple turbine wheel as proposed by Pilbrow. The steam nozzle *b* is situated inside the wheel, and projects steam against the vanes *a*, where its motion is reversed. The fixed vanes *c* lead the steam away. The change of momentum of the steam causes the wheel to rotate.

Fig. 24 shows in side elevation two such wheels mounted on the same shaft and enclosed in the same case. The vanes are set opposite ways on the two wheels, one wheel being intended for giving a reverse motion to the shaft. The pipes

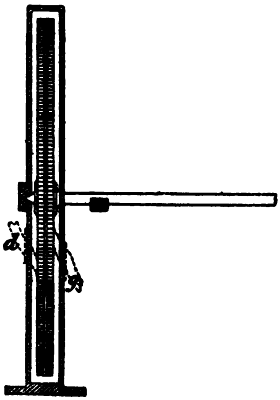


FIG. 24.—Reversing Turbine of Pilbrow's.

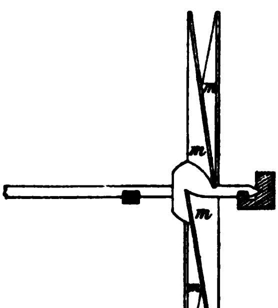


FIG. 25.—Pilbrow's Air-propeller.

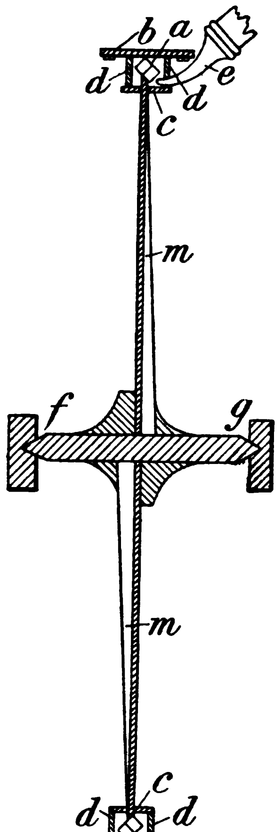


FIG 26.—Combined Steam Turbine and Air-propeller.

conducting the steam to the two nozzles are shown in dotted lines and lettered *d* and *g*. Of course only one wheel and one nozzle are used at a time.

For purposes of land locomotion the inventor proposes to use an air-propeller, as shown in Fig. 25, fixed to the shaft of the steam turbine. Fig. 26 shows in section a combined steam turbine wheel and air-propeller. *mm* are the propeller blades, such as those seen in Fig. 25, and *f*, *g* is the axle on which the blades are mounted. A rim, *c*, is attached to the tips of the blades, and revolves close to the edges of the annular plates *d*, which, with the hoop *b*, form

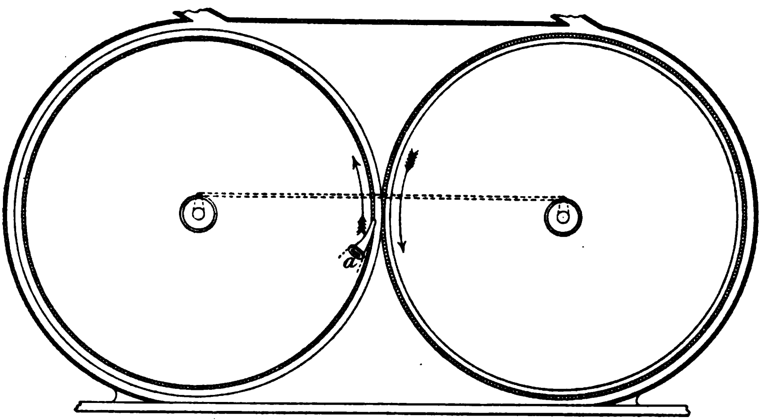


FIG. 27.—Pilbrow's Successive-expansion Turbine: Elevation.

an annular gutter. Inside this gutter, and attached to the rim *c*, are the vanes *a*, which are acted on by the steam issuing from the nozzle *e*. An eduction pipe may be provided to lead the exhaust steam away from the gutter, or this steam may be allowed to escape only at the annular openings between the fixed plates *d* and the revolving rim *c*.

In order to get a steam turbine to work efficiently at a lower speed, the inventor proposes the arrangement shown in Figs. 27 and 28. A number of wheels are placed to

rotate on two parallel axes, the rims of the wheels overlapping, as shown in elevation in Fig. 27 and in part plan in Fig. 28. The wheels are arranged as parallel-flow turbines, and the steam entering the first wheel from the nozzle *a*, passes in succession through the vanes of all the wheels.

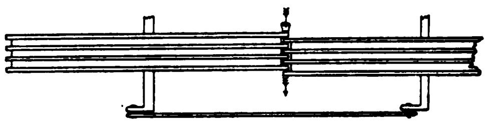


Fig. 28.—Pilbrow's Successive-expansion Turbine: Plan.

This is illustrated as regards two of the wheels by Fig. 29, which is drawn to a large scale. It will be seen that, at the parts adjacent to the nozzle, the vanes of the two sets of wheels move in opposite directions—that is, the two sets of

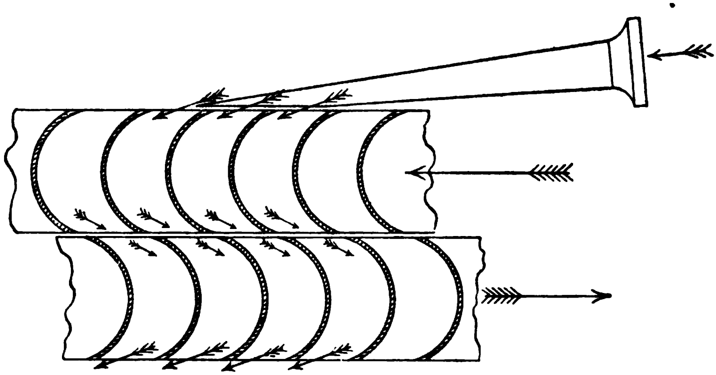


Fig. 29.—Pilbrow's Successive-expansion Turbine: Nozzle and Vanes.

wheels have similar angular velocities. The two axes may be connected by cranks and coupling-rods.

The inventor also apparently conceived the idea of reducing the vane velocity without the necessity of a second shaft by using fixed vanes or guides, for he says, "I also claim the exclusive use of curves or cavities in a stationary case to

reflect the steam back upon the wheel for a second or other number of impulses."

The inventor further describes how the power of one of his turbine wheels may be communicated to machinery by friction gearing.

The most important part of this specification is, in the author's opinion, the description of the method of reducing the vane velocity without losing efficiency by passing the steam through a number of rings of vanes in series. The adoption of this principle in the Parsons turbine has contributed much to make the latter so serviceable.

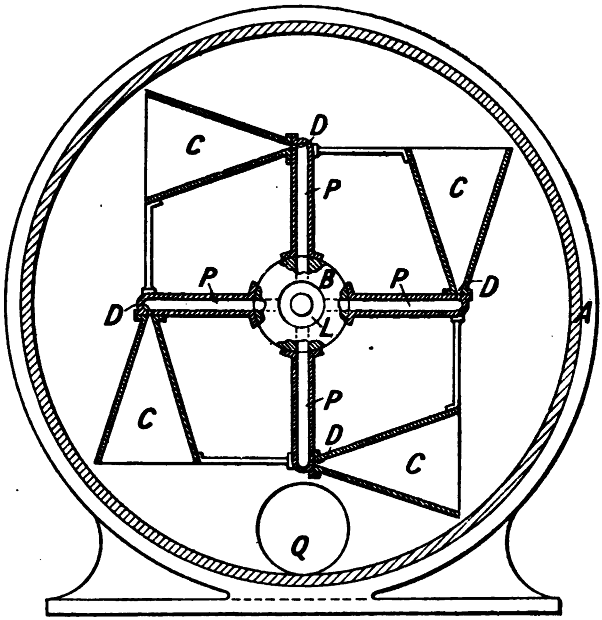


FIG. 30.—Von Rathen's Turbine.

Von Rathen's specification, No. 11,800 of 1847, contains descriptions of several varieties of rotary steam or air engines, some at least of which may be classified as turbines. Fig. 30

shows in section one variety. A is a fixed casing in which rotates the boss B, carrying the radial pipes P. At the end of each pipe P is a cone, C, whose smaller end communicates with the interior of the pipe by means of a small orifice, D. Steam is supplied to the pipes P through the hollow boss L, and escapes, after expansion in the cones, by the pipe Q, to the atmosphere or the condenser. The boss B is mounted

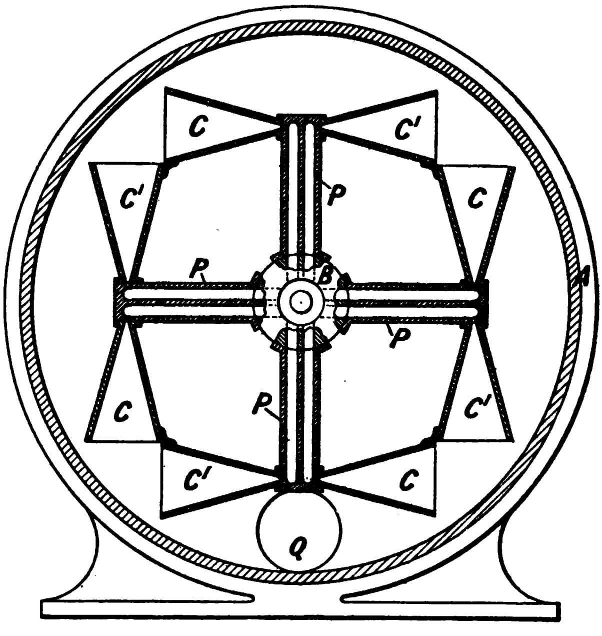


FIG. 31.—Von Rathen's Reversing Turbine.

on an axle, which passes through the flat sides or ends of the casing. To render these parts steam-tight, the inventor proposes to use metallic bushes or packings, "and rings of gutta-percha, sulphurized caoutchouc, or similar substances." Fig. 31 shows a modification of the type of engine just mentioned intended for reversing. The pipes P are here made double. One chamber of each pipe communicates with a cone,

C, while the other chamber communicates with a pipe, C′. Steam can be admitted either to the cones C or the cones C′, and the engine can, therefore, rotate in either direction. The

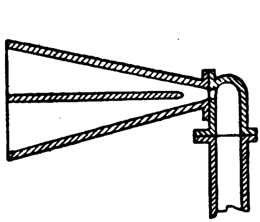


FIG. 32.

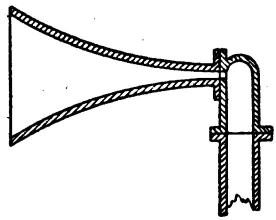


FIG. 33.

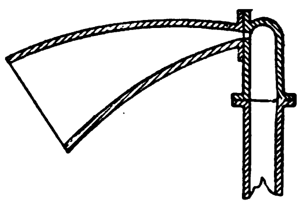


FIG. 34.

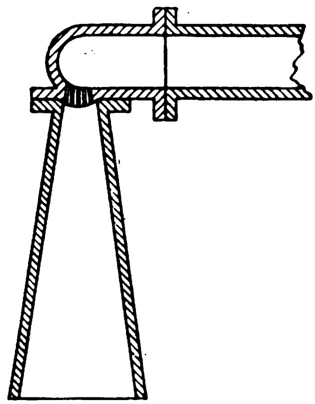


FIG. 35.
Forms of Expanding Cone or Nozzle for Von Rathen's Turbine.

inventor describes and illustrates various constructions of expanding cones or their equivalents. Some of these are illustrated in Figs. 32, 33, 34, and 35. Several other varieties of engine are described, in some of which the casing revolves as well as the boss.

In 1848 **Robert Wilson,** of Greenock, was granted a patent for improvements relating to rotatory engines. His improvements are chiefly with regard to the successive expansion of the steam. Wilson states in his specification that he is aware that, previous to his invention, steam has been employed in reciprocating engines to act successively in two cylinders, but that rotatory reacting engines have hitherto been worked only

so as to utilize the force of the steam at a single operation. The last part of the statement is not correct. Wilson seems to have been unaware of Pilbrow's compound steam turbine. But although Wilson's invention does not contain all the novelty that he attributed to it, it is nevertheless very interesting, and the specification shows that the inventor had carefully considered all the details of his engines. Some of his forms

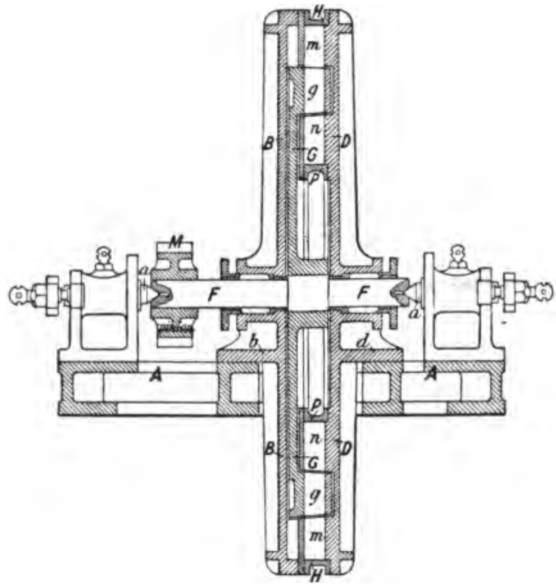


Fig. 36.—Wilson's Radial-flow Turbine with Single Ring of Moving Blades: Sectional side elevation.

and methods of construction have just recently been put in practice very much as he proposed.

One form of Wilson's turbine is shown in Fig. 36, in sectional side elevation; while Fig. 37 shows the same, half in front elevation and half in section. On a base plate, A, are mounted two discs, B and D, which are united at their circumferences by the ring H. Each of the discs has a stuffing-

box through which passes a shaft, F, adapted to rotate on conical pins, *a*. On the shaft, and between the discs B and D, is keyed a disc, G, and this disc carries a number of curved vanes, *g*, which are best seen in Fig. 37. The disc D carries a number of vanes, r^1, r^2, r^3, etc., and also (presumably) a number of blocks, M, separating chambers m^2, m^3, m^4, etc.

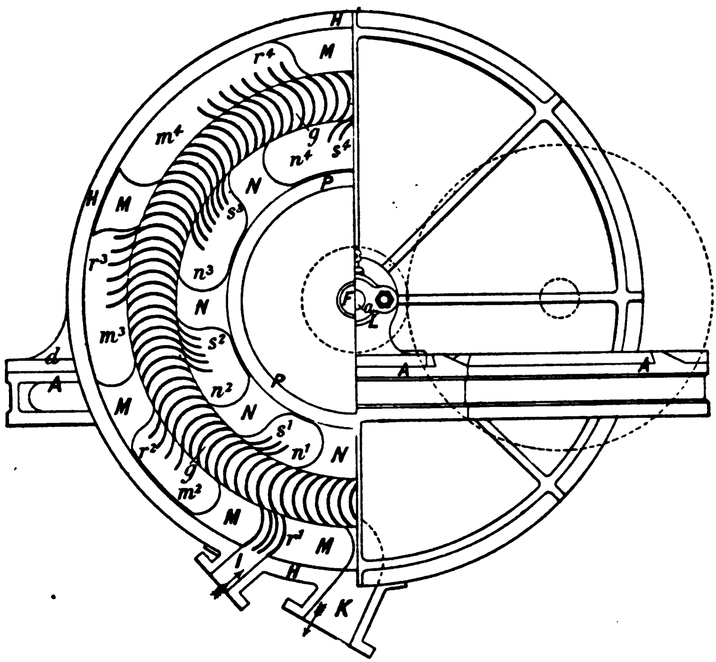


Fig. 37.—Wilson's Radial-flow Turbine with Single Ring of Moving Blades: Half section and half front elevation.

(lettered *m* in Fig. 36). The disc D also carries a number of vanes, s^1, s^2, s^3, etc., and (presumably) a number of blocks, N, separating chambers n^1, n^2, n^3, etc. (lettered *n* in Fig. 36). All the vanes are arranged in three concentric rings so that steam can pass (for example) through between the vanes r^1 and *g*, or (for example) from the chamber m^2, through between

the vanes s^1 and g to the chamber n^2, without any movement parallel to the axis of revolution of the shaft F. This is shown clearly in Fig. 36. The steam passes through the ring H at I, and between the vanes r^1 which guide it to strike the vanes g nearly tangentially to these. The steam passes through between the vanes g, enters the chamber n^1, sweeps round this chamber, and re-enters the spaces between the vanes g by way of the fixed vanes s^1. The steam then enters the chamber m^2, sweeps round it, and again enters the spaces between the rotating blades by way of the fixed blades r^2. The steam thus proceeds round the casing with a serpentine course, and eventually leaves the casing at K. The actual path of the steam will be somewhat as indicated in Fig. 38, where the solid line represents the path of the steam, and the dotted lines the internal and external peripheries of the ring of moving vanes. The stream of fluid will of course spread out in its path. The disc G, with its vanes g, presumably moves at a much lower speed than the velocity of rotation of the steam round the axis of rotation of the disc. The multiple action of the steam thus allows nearly all the energy of the steam to be conveniently used, and allows of the rotation of the

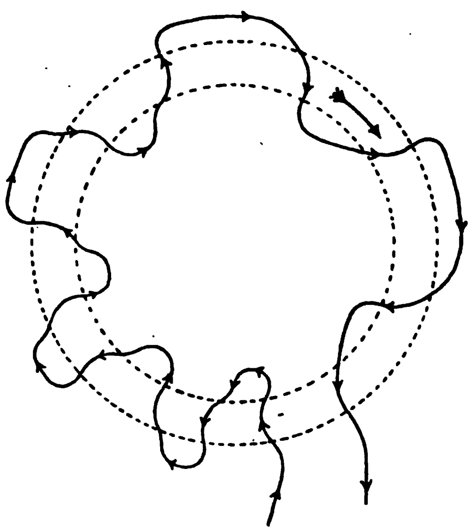


Fig. 38.

moving vanes at a speed which is small compared with the absolute velocity of the steam.

Fig. 39 shows another form of Wilson's turbine, in which the rings of blades *v*, *t*, and *s* are attached to a disc keyed on a revolving shaft, while the vanes *w*, *u*, and *g* are attached to a disc which is either stationary or is keyed

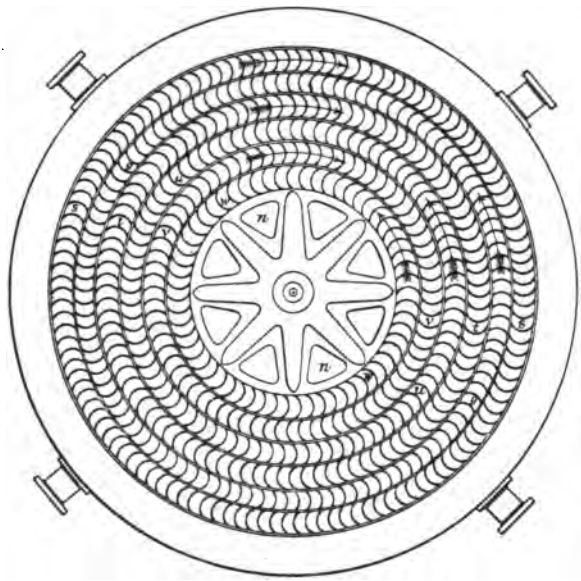


FIG. 39.—Wilson's Radial-flow Turbine with a series of Rings of Moving Blades.

to a shaft revolving in the opposite direction to the first-mentioned shaft. Steam is supplied from the boiler to the space *nn*, enters at several points the spaces between the blades, and works its way outwards through all the rings of blades. Fig. 40 shows a third form of Wilson's turbine, in which the blades *g*, *u*, and *w* are attached to and revolve with the shaft F, while the blades *v*, *t*, and *s* are fixed to the casing H, and do not move. The last two forms of Wilson's turbine are improvements on Pilbrow's device for obtaining

multiple action of the steam, and are the same in principle as successful turbines of the present day. Wilson's turbines were not intended to be mere toys. One of them is shown in the specification drawings as over 9 feet in diameter.

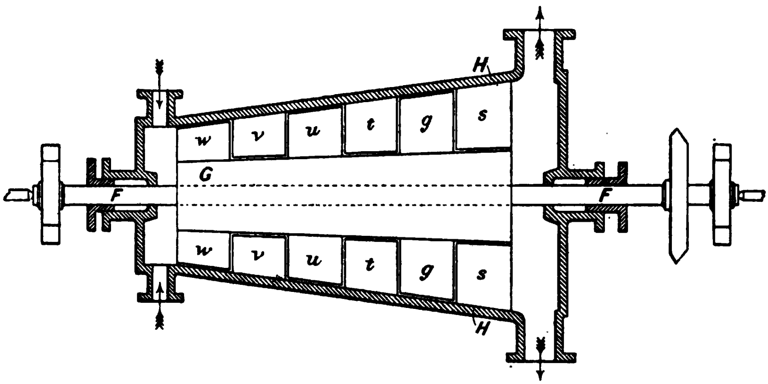


FIG. 40.—Wilson's Parallel-flow Turbine.

Fernihough's Patent, No. 13,281 of 1850. The patentee describes an apparatus in which the products of combustion from a furnace mingled with steam or water-spray are used to drive a turbine.

In 1853 the French mining engineer **Tournaire** pointed out very clearly the requisites of a successful steam turbine. Tournaire explained that elastic fluids like steam acquire enormous velocities, and that in order to properly utilize these velocities in a simple wheel, the latter would require to have an extraordinary great speed. He further explained that the difficulty of excessive speed of rotation could be avoided by causing the steam or gas to lose its pressure in a gradual manner, or by successive fractions, and by making it act in series on a number of turbine blades. Tournaire described a machine in which there were several shafts, all of which carried pinions which geared with a common shaft

from which power could be taken. Each shaft carried a number of wheels with blades, which wheels alternated with a number of rings of blades fixed to an enclosing cylinder. The steam, after passing in series through the fixed and moving rings of blades in one cylinder, was led to the cylinder enclosing the second shaft, and so on. Tournaire recognized that very good workmanship would be required to prevent serious loss of power through leakage between the fixed and moving blades. He also recognized the difficulty with toothed wheels rotating at the necessary speeds, and suggested the use of helicoidal gearing.

The good workmanship referred to by Tournaire has contributed largely to the success of the Parsons turbine, while the helicoidal gearing is an important feature of the De Laval motor.

Patent No. 3161 of 1873, **Thomas Baldwin.** This inventor, who filed no drawings with his specification, proposed to use a machine in the form of an hydraulic turbine, in which the flow of the steam might be "inward, or outward, or parallel." He mentions that a disc may be caused to rotate by the reaction of steam-jets issuing from apertures at its periphery, or by the impulse on the disc of steam-jets issuing from apertures in the casing. The inventor proposes to employ several machines in series, the steam which exhausts from the first being employed to drive the second and then the others in succession. It is proposed that the action of the steam on the last machine should be increased by leading it therefrom to an injector or ejector where the steam would be condensed, and the kinetic energy of the condensing water would then be utilized in a hydraulic turbine or water-wheel.

Patent No. 706 of 1874, **Alexander Teulon.** This inventor

proposed to utilize the axial thrust of a steam turbine to balance the axial thrust of a screw propeller.

Figs. 41 to 46 show steam turbine details which formed the subject-matter of several letters patent granted to **John S. Raworth**, about 1894.* 1, 1^a, 1^b, Fig. 41, are ports in communication with the nozzles of a turbine, and 2 is a circular valve furnished with ports, 2^a, 2^b, 2^c, in the form of slots with circular ends. The governor is connected to the valve,

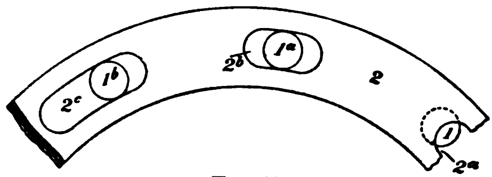


Fig. 41.

so that, when the load on the turbine falls, the valve is turned to the right, and cuts off the steam supply, first to the port 1, and then in succession to the ports, 1^a and 1^b. When the load is increased, the valve is caused to move in the opposite direction.

Fig. 42 shows a compound nozzle, which is intended to be screwed at 3 into the main steam duct. The jet of steam flowing from the main steam-duct commences to expand at 4, and, as the steam increases

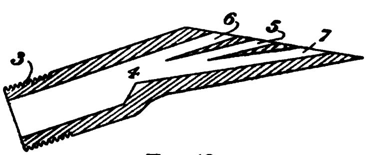


Fig. 42.

in velocity, the nozzle is developed into two or more parts, 5, 6, 7.

Figs. 43 and 44 show a device or arrangement for reducing the high speed of steam turbines by gearing to a speed suitable for ordinary industrial purposes. The turbine shaft 1 is supported in a bearing, 2, and carries a small friction

* No. 25,090, dated December 30, 1893; No. 84, dated January 2, 1894; and No. 1242, dated January 19, 1894.

wheel, 3, which gears with large friction wheels 3^a and 3^b. These large wheels are mounted on shafts, 4 and 4^a, which carry toothed pinions, 9^a and 9^b, which gear with a spur-wheel, 9, mounted on a shaft, 10, from which power can be taken. The shafts 4 and 4^a are supported in bearings in levers, 5^a and 5^b, which are pivoted at 6 and 6^a to the base-plate 7, and are linked together at their upper ends by the rod 8, having a head, 8^a, and a nut, 8^b. A spring, 8^c, is arranged on the rod so that, by adjusting the nut 8^b, the wheels 3^a and 3^b can be pressed against the small wheel 3 with any desired pressure.

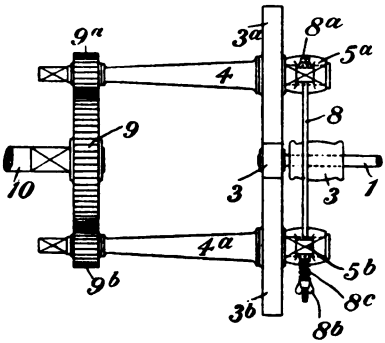


Fig. 43.

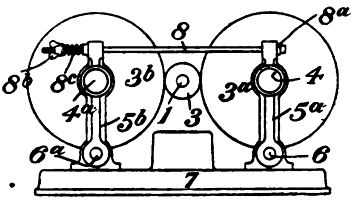


Fig. 44.

Fig. 45 shows another method of reducing the speed. The turbine shaft a' carries a pulley, a, which gears frictionally with three wheels, b, of which only one is shown. The wheels b rotate on studs, f, attached to swing-frames, g, one of which is shown separately in Fig. 46. Each wheel, b, is lubricated by means

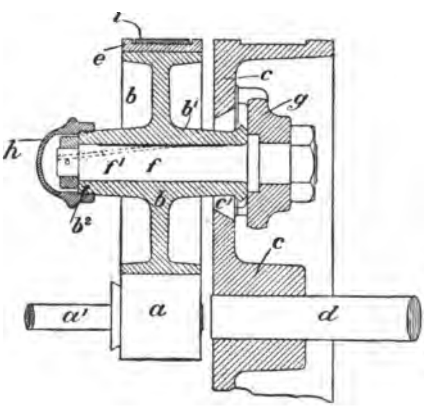


Fig. 45.

of a channel, f', leading from an oil-chamber enclosed by the cap h screwed on the boss b^2 of the wheel. This construction prevents oil dripping on to the friction wheels. The frames g are pivoted at g' to the plate c, to which is keyed the power-shaft d. The frames may be weighted at g^2 to balance the weights of the studs and friction wheels. The latter are pressed against the small wheel a by a flexible band, e, which encircles the three wheels b, and is of such a diameter that it has to be sprung to extend around them. The band may be prevented from rotating by a band-brake, i.

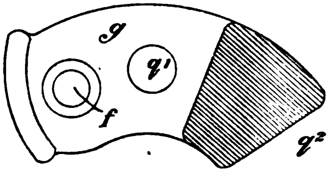


FIG. 46.

Alexander Morton, of Glasgow, made several experiments with steam turbines about 1888 to 1892. In one of his engines a series of cylinders was arranged one within the other, the ends of the whole being closed by two common discs. Steam was admitted to the interior of the inner cylinder, and expanded through nozzles into the surrounding cylinder, and this action was continued till the steam reached the last cylinder, which was in communication with a condenser. This action of the steam caused the cylinders to rotate, all moving together. No guides whatever were used during the several stages of expansion, and the engine acted wholly by reaction. Parts of three of the concentric cylinders are shown diagrammatically in Fig. 47, the nozzles also being shown. The large arrow indicates the direction of rotation of the cylinders, and the small arrows the direction of motion of the steam relatively to the cylinders.

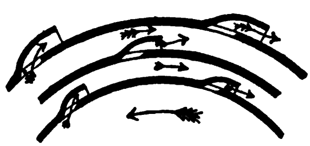


FIG. 47.—Concentric Cylinders and Nozzles of Outward-flow Turbine of Morton's.

In another of Morton's engines (proposed, if not tried) the steam was conducted from the centre of a rotating part to the circumference by way of a number of converging channels, and was then allowed to expand in a tangential direction through

FIG. 48.—Steam Duct and Nozzle of Outward-flow Turbine of Morton's.

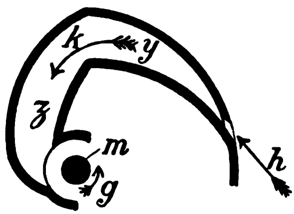


FIG. 49.—Steam Passages for Inward-flow Turbine of Morton's.

a number of diverging nozzles. Fig. 48 shows the construction diagrammatically, one converging passage, *a*, and one diverging nozzle, *b*, being shown; *c* represents the shaft which carries and is driven by the rotating parts; the arrow *d* represents the direction of rotation of this shaft; and the arrows *e*, *f* represent the direction of flow of the steam in the channel and nozzle.

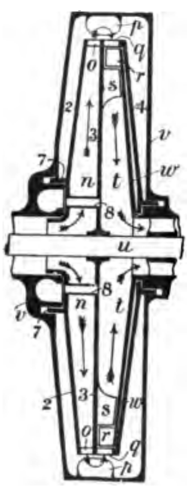


FIG. 50.—Inward and Outward-flow Turbine of Morton's.

Fig. 49 indicates diagrammatically the arrangement and form of passages, *y*, *z*, for an inward-flow turbine, the arrow *g* showing the direction of rotation, and the arrows *h* and *k* the direction of flow of the steam relatively to the rotating parts; *m* is the axis of rotation.

Fig. 50 illustrates diagrammatically part of a radial-flow turbine of Morton's, in which the steam alternately passes inwards and outwards. The arrows indicate the path of the steam, which flows freely from the centre of the rotating conical chamber *n* to the periphery of the same,

where it passes through divergent passages, *o*, of the nature of that shown at *b* in Fig. 48. It then has its motion changed by guides, *p*, and traverses divergent passages, *q*, somewhat similar to that shown at *y* in Fig. 49. The steam, continually expanding, has its motion then altered by guide-vanes, *r*, and impinges on rotating vanes, *s*. It then passes to the centre of the conical chamber *t*, where it escapes from the turbine, or is again similarly treated. The passages *o* and *q* and vanes *s* are arranged so as all to help to rotate the chambers *n* and *t* and the shaft *u*. The casing *v* is fixed, as is also the dished plate *w*, which supports the guide-vanes *r*.

The steam in the chamber *n* will press with equal intensity on the plate 2 as on the plate 3; but the steam in the chamber *t* will not press with equal intensity on the plate 3 as on the plate 4, if the fixed plate *w* be made solid. Further, there is no portion of the plate 2, and no portion of the plate 4, corresponding to the central portion of the plate 3; and, as this centre portion of the plate 3 has unequal pressures on its two sides (for the steam expands in passing from the chamber *n* to the chamber *t*), there will be a net axial pressure from left to right.

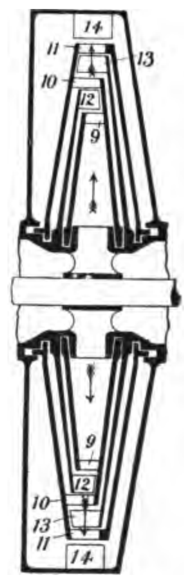


FIG. 51.—Arrangement of Vanes and Channels in Morton's Turbine.

This axial pressure is balanced by shutting off a portion of the exterior of the plate 2 from the pressure in the casing *v* by means of the ring 7, the part of the plate within the ring being subjected to the pressure in the chamber *t* by means of the tubes 8.

Another arrangement of vanes and channels is shown

diagrammatically in Fig. 51, the steam passing radially outwards as indicated by the arrows, and traversing in succession diverging passages 9, 10, 11. Guide-vanes 12, 13, and 14 receive the steam after leaving the diverging passages, and redirect its course.

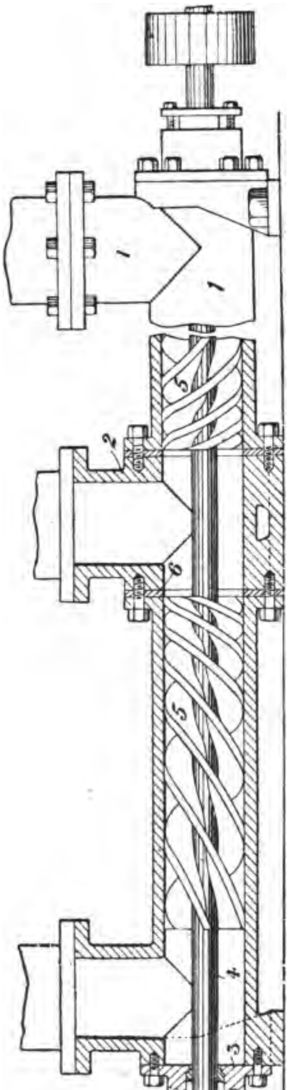


Fig. 52.—Screw Type of Steam Turbine.

Fig. 52 shows in partial sectional elevation a steam turbine of the screw type, experimented on by **Professor Hewitt.** A shaft 4 is provided in a cylindrical casing, in the ends of which are stuffing-boxes. The shaft is provided with screw-threads, 5, whose pitch increases from the centre to the ends. Steam or other fluid enters the casing by way of the branch 2, and, passing through holes in the plates 6, gains access to the helical grooves between the screw-threads. The steam leaves the casing by the branches 1 at the two ends. One of the plates 6 is shown separately in Fig. 53. Professor Hewitt states that this turbine did not give good results, and that he considers that this was due to the absence of guide-plates for the steam. This is probably the case. The steam

would, no doubt, act effectively when it first struck the screw-threads; but, after it had once been deflected into a helical course, it would rush to the exhaust port, without producing much additional effect as regards rotating the shaft.

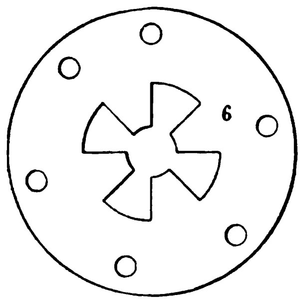


FIG. 53.—Admission Plate.

Only a small selection of inventions relating to the steam turbine could be reviewed in this chapter. Several not here referred to are described in the paper presented by Mr. Sosnowski to the International Congress on Applied Mechanics, held in Paris in the summer of 1900.

CHAPTER III.

HISTORY OF THE PARSONS STEAM TURBINE.

On April 23, 1884, the Honourable Charles Algernon Parsons filed two applications for letters patent. These were the first patents of the great inventor relative to steam turbines, although he had previously experimented with rotary engines of another type. One of these patents is entitled "Improvements in Rotary Motors actuated by elastic fluid pressure, etc." An engineer reading this specification is at once struck with the apparent practicability of the motor therein described compared with most of its predecessors of a similar type. The motor as described and illustrated shows that an immense amount of thought and attention had been spent on details —on devices for reducing cost of construction, for preventing vibration, for drawing off leaking steam, for providing efficient lubrication, etc. This attention to details has characterized the Parsons turbine throughout its life (short as yet), and probably to this is largely due the immense success of the present-day motor.

No attempt will be here made to describe in full the first Parsons turbine, as some of the details are now obsolete, but some of its interesting features are here illustrated and explained. Fig. 58 is a plan, partly in section, of the main part of the motor. A spindle, S, is formed with a central

collar, S^1, and reduced ends, S^3. On S are placed a number of rings, B, B, which are held in place between the collar S^1 and nuts S^2 screwed on the spindle. The rings are provided at their circumferences with blades, b, b^1, b^2, which are interspaced between blades, f, f^1, f^2, fixed in the inside of the turbine casing. Steam is admitted to the annular chamber g, and passed through the rings of blades in series till it reaches the exhaust ports h, h. Any steam that leaks through to the annular chambers, o, o, is led away to a chamber, P (Fig 59), where by the action of a live steam-jet issuing from the nozzle p, it is ejected through the pipe q. As the steam passes from the centre to both ends, there can be little axial thrust on the shaft, but what little does occur is balanced by the exhaust steam at the ends of the casing, the arrangement being such that a slight

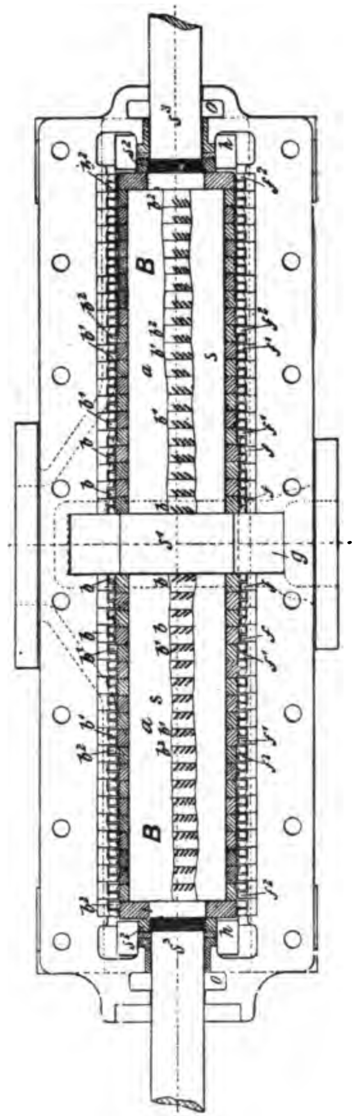


FIG. 58.—Early Parsons Turbine.

movement of the shaft to either end of the casing checks the

exhaust at that end, and so increases the back pressure. In order that the shaft and rotating parts may rotate about their centre of gravity instead of about their geometric centre when the two are not coincident, arrangements are provided for allowing the shaft a little lateral play. One of these arrangements is shown in Fig. 60, where I is a light bush enclosing

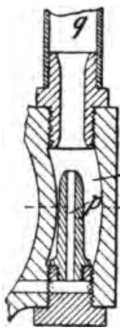


FIG. 59.—Escaped-Steam Ejector.

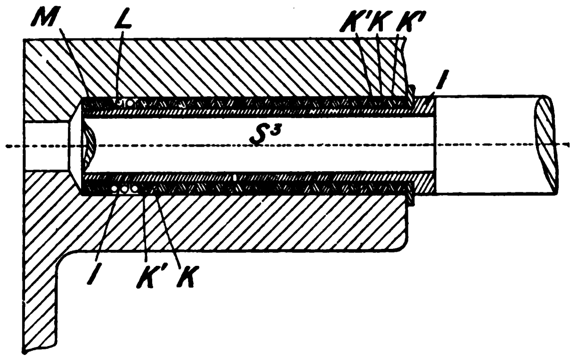


FIG. 60.—Bearing for Spindle in Early Parsons Turbine.

the shaft. Surrounding this bush are rings, K, which touch the casing but not the bush, alternating with rings, K', which touch the bush but not the casing. The nut M compresses the spiral ring L against the end ring K'. The shaft can thus move laterally a certain extent, say, one-hundredth of an inch, but this movement is resisted by the friction of the collars on one another. A system of forced lubrication is provided, and also a fan governor.

A steam turbine dynamo was constructed in 1885 by Messrs. Clarke, Chapman, Parsons and Co. Revolving at the rate of 18,000 revolutions per minute, it gave great satisfaction, and was used for several years generating current for incandescent electric lamp manufacture.

A year or two later Parsons introduced an improved

steam turbine, of which an elevation, partly in section, is given in Fig. 61. The steam entered at *a*, and passed through the rings of blades shown diagrammatically at *c* and *c'*. The fluid then passed through the rings of blades of larger diameter indicated by the letters *e* and *e'*, and then through those of still greater diameter situated at *g* and *g'*. The exhaust ends of the parts *c* and *c'* were connected by the passage *d*, which maintained an equal pressure at the two points, and the exhaust ends of the parts *e* and *e'* were similarly united by the passage *f*. The exhaust from this compound turbine was taken away from both ends by the passage *h*. Water or steam packing was provided at the places where the spindle passed through the ends of the casing, so that water or steam might be drawn into the condenser, but no air could. An annular chamber, *i* (Figs. 62 and 63), was provided round the spindle *b*,

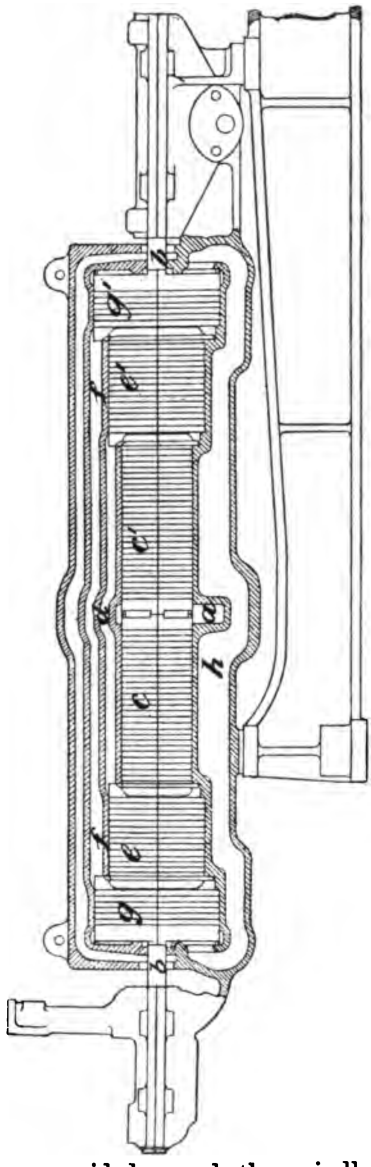
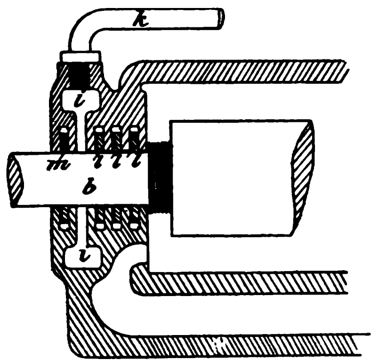


FIG. 61.—Double-ended Parsons Turbine of Increasing Diameter.

and kept supplied by the pipe *k* with water from the hot well or with steam, either at boiler pressure or partly expanded. Packing rings, *l*, *l*, *m*, were used, as shown in Fig. 62, or, when water was employed, the spindle was sometimes cut with right- and left-hand threads, as shown in Fig. 63, so that its rotation tended to repel the water leaking past.



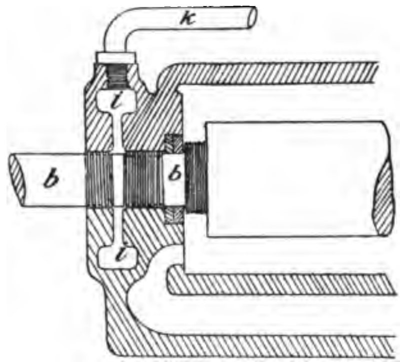


Fig. 62. Fig. 63.

Steam or Water-packing for Spindle of Parsons Turbine.

In 1891 the first Parsons condensing steam turbine was constructed for the Cambridge Electric Supply Company by the firm of C. A. Parsons and Co., just then formed (Messrs. Clarke, Chapman, Parsons and Co. having dissolved partnership in 1889). This engine was tested by Professor Ewing, and its efficiency proved to be equal to that of the best reciprocating engines of the same power.

This condensing steam turbine was followed by many others, plants being supplied to the Newcastle and District Electric Lighting Company, the Cambridge Electric Supply Company, and the Scarborough Electric Supply Company. At first the turbines had all been comparatively small, but larger machines were now made, and the increase in size,

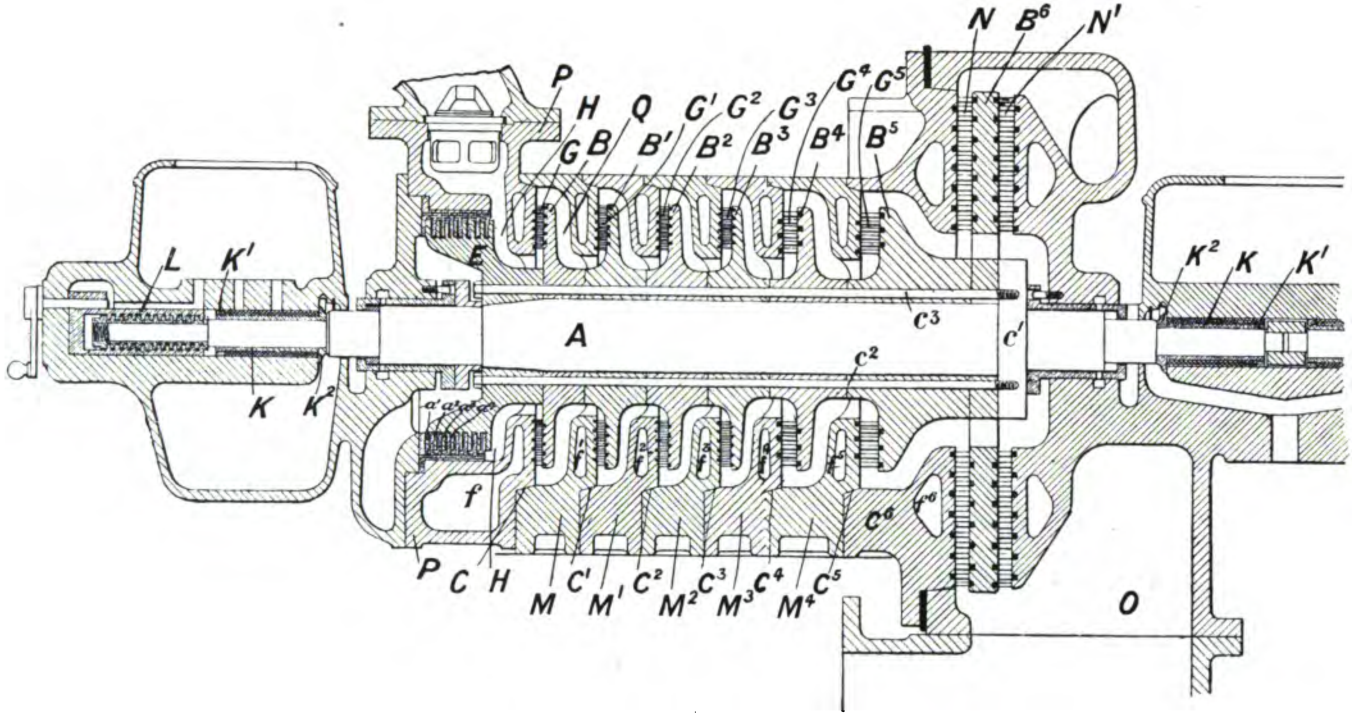


Fig. 64.—Section of Parsons Radial-flow Turbine.

together with improvements in design, led to still higher efficiencies.

Fig. 64 shows a recent form of construction of radial-flow Parsons turbine. Steam is led into the annular chamber H, and passed therefrom through the fixed and moving rings of blades G, of which the fixed blades are attached to the casting P, and the moving ones to the disc B. The steam, in a somewhat expanded state, then doubles back along the passage Q, and works its way outwards again through the rings of blades G^1. The fixed blades in this case are attached to the annulus M. The action is repeated through the rings of blades G^2, G^3, G^4, and G^5. The form of these rings of blades is shown in Figs. 3–10, pp. 3–6. The final expansion of the steam takes place in the rings of blades N and N^1, and the steam then reaches the passage O and proceeds to the condenser. The method of fitting the casting P to the parts M, M^1, M^2, etc., by means of spigot and faucet joints, is clearly shown.

E is a balance piston used to balance the end pressure of the steam on the discs B, B^1, B^2, etc. This piston is provided with deep projecting flanges, a, a^1, a^2, a^3 (Figs. 64 and 65), which flanges are adapted to rotate in corresponding recesses provided in a ring secured to the casting P. The flanges are serrated on one side, as shown at b, b^1, b^2, and b^3. The resistance to the flow of steam through the tortuous passages between the fixed and moving flanges is very great, and leakage is thus reduced to a minimum. The piston E is mounted on a conical part of the spindle.

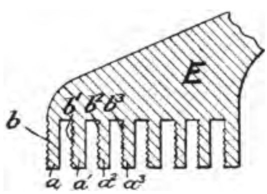


Fig. 65.

The turbine spindle A is constructed with a collar, c^1, into which are screwed long studs or pins, c^2, c^3, which pass

through holes in the turbine discs B, B^1, B^2, etc., and through holes in the balance piston E. The discs and balance piston are thus firmly held on the spindle. Live steam is admitted to the annular spaces f, f^1, f^2, etc., to reduce the condensation of the steam passing through the rings of blades.

In order to damp vibration and to allow the spindle a little transverse movement so that it may rotate about the line containing the centre of gravity of the revolving parts, the spindle is enclosed near both ends in a sleeve, K (Figs. 64, 66, 67), provided with a flange, K^2, and a collar, K^1.

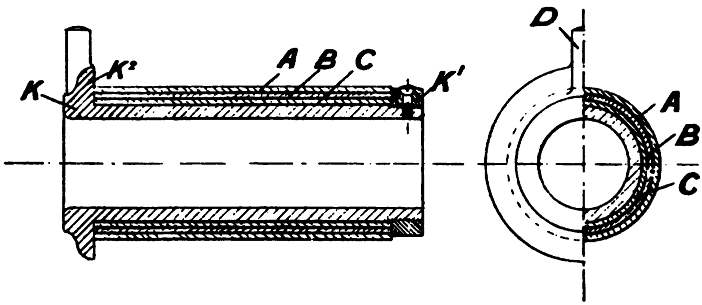


FIG. 66. FIG. 67.

Bearing for Spindle of Parsons Turbine.

Surrounding the sleeve, and between the flange and collar, are placed three concentric tubes, A, B, and C. The tubes are bored so as to be an easy fit on each other and on the sleeve; and oil is supplied to the thin annular spaces so formed so that any transverse movement of the shaft is resisted by the fluid friction of the thin films of oil which have to be squeezed from the parts where the tubes are compressed against each other. Figs. 68 and 69 show an alternative construction, where two tubes, A and E, contain between them several segments, F, G, H, which are cut from a tube of smaller diameter so that the ends of the segments touch the

inner tube E, and the middle portions of the segments touch the outer tube A. Oil is supplied in this case also to the

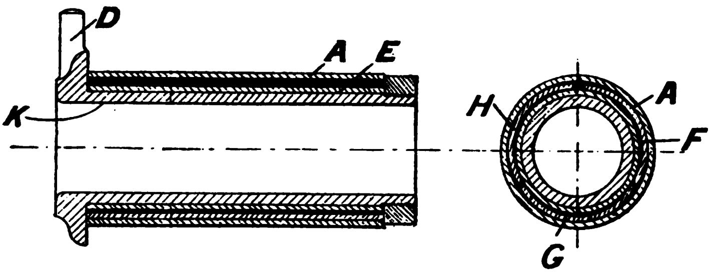


FIG. 68. FIG. 69.
Elastic Bearing for Parsons Turbine.

spaces between the tubes and sleeve, but the fluid friction is aided by the elasticity of the segments F, G, H. In both cases suitable means, such as projections D, are provided to prevent rotation of the sleeve K.

The end-thrust of the spindle due to the pressure of the

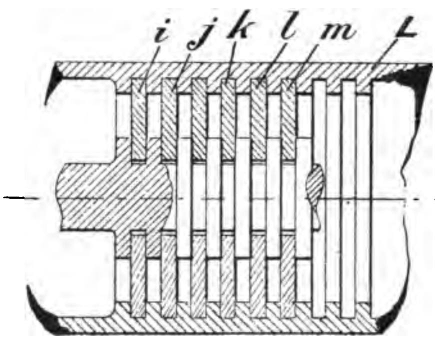


FIG. 70.—Thrust-block of Parsons Turbine.

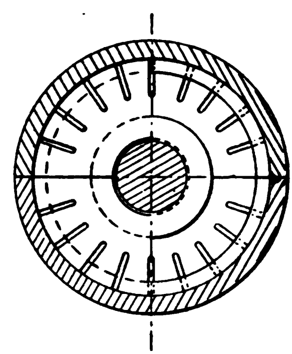


FIG. 71.—Slotted Ring for Thrust-block.

steam on the discs B, B^1, B^2, etc., is taken up by the thrust-block L (Fig. 64), which is made in halves and provided with flanges and recesses to engage with recesses and flanges on the spindle. Sometimes the construction shown in Fig. 70 is

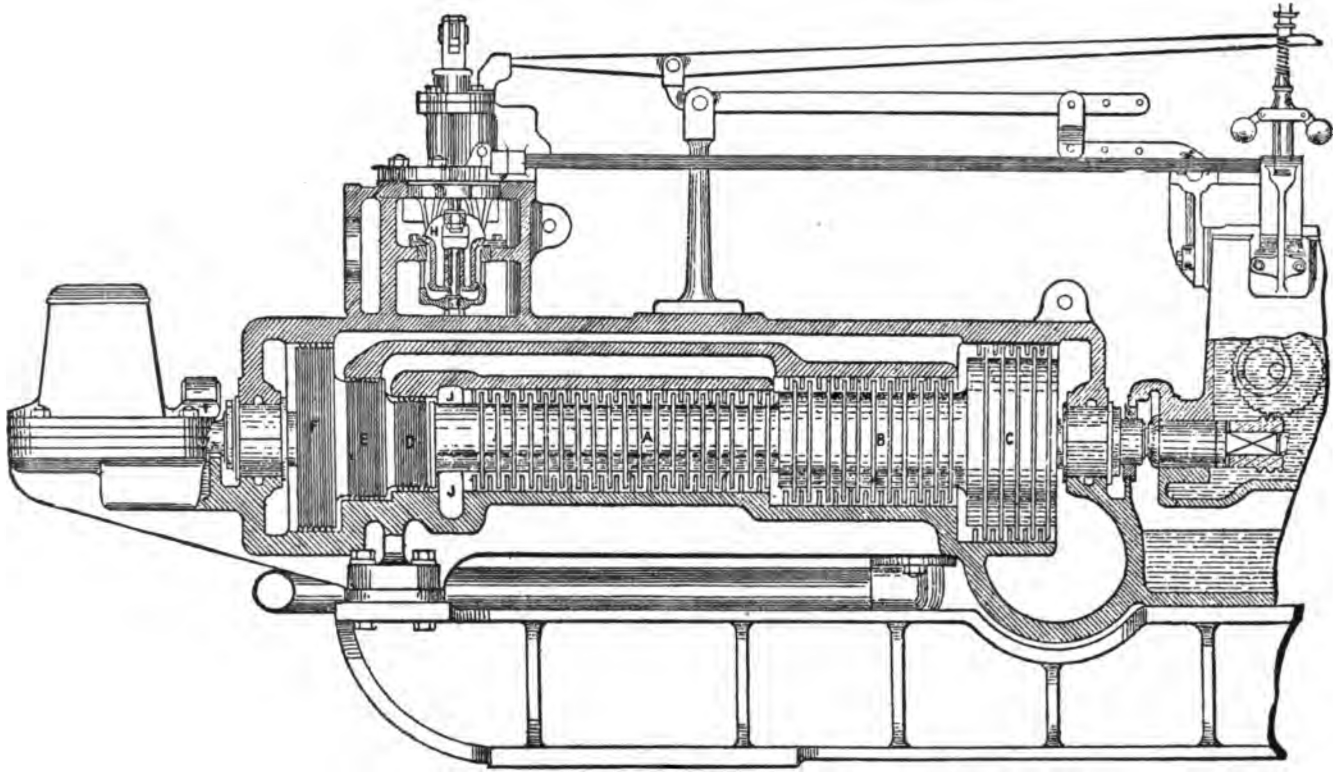


FIG. 72.—Section of Parsons Parallel-flow Turbine.

adopted where rings, *i*, *j*, *k*, *l*, *m*, are used, which are separate from both block and spindle, and are of sufficient diameter and thickness to possess the requisite elasticity. The elasticity may be increased by providing slots in the rings as shown in Fig. 71; or spring washers may be inserted between the rings and the recesses for them in the block L.

All these devices for taking up end-thrust and damping vibration have been patented by Parsons.

Fig. 72 shows in vertical longitudinal section a modern Parsons parallel-flow turbine. Steam passes through the

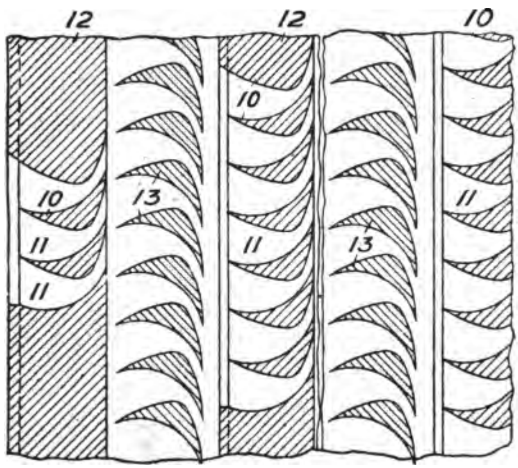


FIG. 72A.—Fixed and Moving Blades of Parsons Turbine.

equilibrium valve H and enters the annular space J, from which it proceeds through the fixed and moving blades in the high-pressure cylinder, or part A; then through those in the intermediate cylinder, or part B; and then through those in the low-pressure cylinder, or part C. The arrangement and construction of the rings of blades will be best understood by referring back to Figs. 3–8.

Instead of making the turbine cylinder of increasing

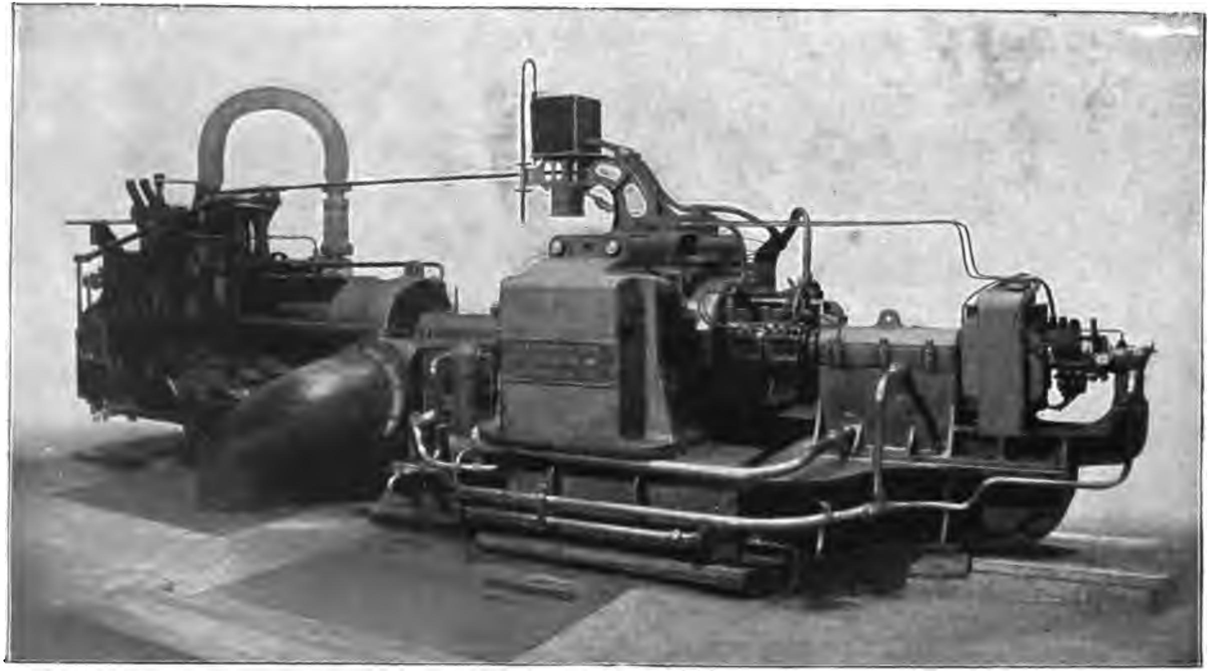


Plate II.—Parsons Steam Turbine coupled to 500-Kilowatt Alternator.

diameter, the fixed rings of blades at the high-pressure end may contain only a few blades, the spaces where blades are not placed being occupied by solid or hollow segments. The number of blades on the fixed rings will then increase progressively from the high-pressure end to the low-pressure end of the turbine. All the moving rings, however, are provided with blades round their whole circumferences. A section of the fixed and moving blades is shown drawn to a large scale in Fig. 72A, where 10 represents the fixed blades, 11 the spaces between them for the passage of steam, 12 the segments occupying the remaining space of the fixed rings, and 13 the rotating blades.

Plate II. shows a Parsons steam turbine coupled direct to a 500-kilowatt alternator. It is installed in the station of the Newcastle and District Electric Lighting Company.

CHAPTER IV.

POINTS OF RESEMBLANCE AND DIFFERENCE BETWEEN THE STEAM TURBINE AND OTHER MOTORS.

THE action of the steam turbine depends on the conversion of the heat energy of the steam into kinetic energy, and then in the transference of this kinetic energy from the steam to the rotating parts of the turbine. The latter part of the action is thus in principle much the same as that of the water turbine, but the former part has no parallel in the hydraulic motor. In a water turbine the fluid is practically at constant volume and at constant temperature, and its kinetic energy is gained at the expense of potential energy due to pressure or position. On the other hand, when steam is used, this fluid varies in volume within very wide limits. Thus, 141 cubic feet of saturated steam at 200 lbs. pressure absolute produces 1647 cubic feet at atmospheric pressure, and this produces only 1 cubic foot of water when condensed. These volumes are represented respectively by the cubes A, B, and C in Fig. 73, p. 52. The temperature of the steam varies also, and care has to be taken to prevent, as far as possible, loss of heat by radiation, a point that does not call for attention with a water turbine.

Another important point of difference between the steam turbine and the water turbine is the immense velocity of

the fluid in the former compared with the latter. In a water turbine working under the large head of 150 feet, the velocity of the fluid entering the wheel is about 96 feet per second.

In steam turbines a fluid velocity of 2000 to 3000 feet per second is common. The reason for high speeds with steam can easily be seen. A cubic foot of water having a velocity of 96 feet per second has a kinetic energy of about 9000 foot-lbs.* A cubic foot of dry saturated steam at 50 lbs. pressure absolute has, however, so small a mass that, in order that it may have the same kinetic energy, it must have a velocity of about 2200 feet per second.* These differences in the physical properties of steam and water necessitate great differences in the construction of steam turbines and water turbines. It should also be noted that all friction in a water turbine means loss of energy; but that in a steam turbine the heat generated by the friction may serve to heat the fluid, and thus in great part restore the energy absorbed. This will be referred to again.

Comparing a steam turbine with a reciprocating engine, we find that, although the greatest possible efficiency, as determined by thermo-dynamic considerations, is the same in both, being represented by Carnot's formula $\frac{T_1 - T_2}{T_1}$, the causes which reduce this efficiency below this maximum are largely different in the two cases. One of the greatest losses in the reciprocating engine is due to the alternate contact of the inside of

* Kinetic energy of 1 cubic foot of water $= \frac{mv^2}{2} = \frac{62\frac{1}{2} \times 96^2}{2 \times 32{\cdot}2} = 9000$ foot-lbs. approximately.

Kinetic energy of 1 cubic foot of dry saturated steam at 50 lbs. pressure absolute $= \frac{mv^2}{2} = \frac{0{\cdot}12 \times 2200}{2 \times 32{\cdot}2} = 9000$ foot-lbs. approximately.

the cylinder with the hot steam and with the comparatively cold exhaust. The cylinder walls rob the entering steam of much of its heat energy. Some of this energy may be recovered by the steam at a later part of the stroke, but a great part is given up to the exhaust, and, unless it can after-

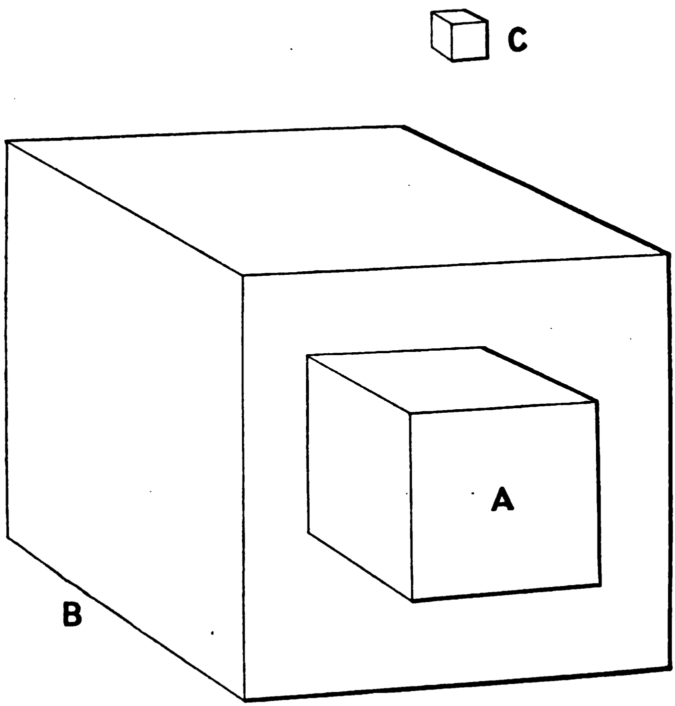


FIG. 73.—A, volume of steam at 200 lbs. absolute; B, volume of steam at atmospheric pressure; C, volume of water.

wards be utilized, is lost. There is no such loss with the steam turbine, as the steam passes constantly in the same direction, some surfaces of the turbine making contact with the entering steam and some with the exhaust, but none with both.

Another loss which is sometimes thought to be considerable occurs with reciprocating engines using slide-valves or their

equivalents, and consists in steam leaking past the valve to the exhaust. This, of course, cannot happen in a steam turbine where no slide-valve or its equivalent exists.

Another great source of loss with reciprocating engines is due to friction. This friction sometimes absorbs more than a quarter of the total I.H.P. of the engine. Except for the friction in the bearings of the shafts, the friction in a steam turbine is of a totally different nature from that in a reciprocating engine. It consists in the friction of the steam against itself and against the surfaces of the turbine, and the friction of the water carried by or deposited by the steam. Water deposited on the fixed parts of the turbine will cause friction by coming in contact with the rotating parts. The amount of clearance between the fixed vanes and the moving vanes of a Parsons turbine is very small, and, with a high speed of rotation, it is quite possible that the friction due to this cause may be considerable. If saturated steam be used, a certain amount must be condensed in the turbine, but, by superheating the steam and jacketing the turbine, this condensation may be very much reduced, if not entirely prevented. It is noteworthy that superheating the steam very much improves the efficiency of a Parsons turbine. Mr. Parsons considers that 55° C. superheat reduces the steam consumption about 12 per cent. It should be borne in mind, however, that a great part of the heat developed by this friction, as already stated, will probably not be lost. Slightly more radiation may take place from the outside surfaces of the turbine, and the exhaust may leave in a slightly less condensed form than would otherwise happen; but a large portion of the heat, it is presumed, will be returned to the working fluid so as to be again utilized. With the reciprocating engine, although the friction of the piston in the cylinder

and of the slide-valve or other valve in the steam-chest may heat the steam, yet, as the exhaust steam receives part of this heat, and as there is much friction caused by other parts than the piston and valve, we may safely assert that in a reciprocating engine the greater part of the heat caused by friction is lost.

Another advantage which the steam turbine possesses over the reciprocating engine is that, with the former, there is no internal lubrication required. The fact that the steam turbine can take steam without any lubricant whatever is doubly advantageous. In the first place, the exhaust steam is absolutely free from oil, so that the water from the hot well can be directly returned to the boiler without the use of an oil filter, and without any danger of the boiler suffering from a deposit of grease in it. The second advantage arises when superheated steam is used. When this is employed in reciprocating engines, there are difficulties with regard to internal lubrication; and there is also the danger of piston and valves sticking, unless properly and carefully designed, owing to difference of expansion of different parts of the engine. With the steam turbine no lubricant is required to be added to the steam, and the danger of harm arising from unequal expansion is not as a rule great. It should be noted that, although both turbines and reciprocating engines improve in efficiency by superheating the steam, the reasons for the superheating are not altogether the same. The reciprocating engine gains chiefly (or at least largely) by the reduction or abolition of initial condensation. This cannot be the chief reason with the steam turbine; but the gain in ecomomy in the steam turbine by superheating will be discussed later on.

The steam turbine benefits more than the reciprocating

engine from a good vacuum in the condenser. (Tables showing the effect of the state of the vacuum on the steam consumption of Parsons turbines are given in Chap. X.) Decreases of pressure below 5 lbs. absolute mean large drops of temperature in the case of saturated steam, and therefore there is a great thermo-dynamic advantage in having a low condenser pressure in a steam-engine. In the case of a reciprocating engine, however, this thermo-dynamic advantage is partly neutralized by the increased initial condensation due to the lower temperatures of the surfaces with which the steam entering the cylinder comes in contact. The increase in efficiency obtained by improving the vacuum is therefore only due to the difference of these two effects. In the case of the steam turbine, however, there is no such initial condensation, and consequently this type of engine gains largely by improvement of the condenser vacuum. Another reason for the comparatively small gain in efficiency by increase of vacuum in the reciprocating engine is the impossibility in most cases of taking full advantage of the vacuum by expanding the steam in the cylinder down to the condenser pressure without unduly increasing the bulk of the engine and diminishing its mechanical efficiency.

A source of loss with the steam turbine which does not occur with the reciprocating engine is caused at the parts where the shaft leaves the case. At high speeds of rotation difficulties obviously occur with packing such as is used in the piston-rod glands of a reciprocating engine. In the Parsons turbine no packing is used, but a special device is employed which will be described hereafter; with this device very little loss is said to occur.

It should be noted in comparing the driving of alternators by steam turbines and by reciprocating engines that, while the

same percentage variation of speed means the same percentage variation of periodicity, a drop (or rise) of, say, 5 revolutions per minute in the one case, does not mean the same variation of periodicity as in the other case; for the number of alternations per revolution in the turbine-driven alternator is, owing to the speed of rotation, less than in the alternator driven by the reciprocating engine, the mean periodicity being the same in both cases.

CHAPTER V.

VANES AND VELOCITIES.

LET us now consider the form of the vanes or blades and the speed of rotation of a steam turbine, and, in the first instance, it may be advisable to deal with turbines generally.

As we shall be using the terms "absolute velocity" and "relative velocity" with respect to the motion of the fluid, it will be better to state here that by absolute velocity is meant a velocity which would be absolute if the turbine casing or frame were at rest. A turbine may be on board a ship, and therefore have the velocity of the ship, and even when on land and what we call fixed, it nevertheless has the velocity of the earth. It is convenient, however, to neglect these velocities of the ship and the earth and such-like, and speak of the velocity of a revolving part of the turbine or of the operating fluid as *absolute*, when we mean that such a velocity would be absolute if the casing or frame, or fixed parts of the turbine, had no motion. We shall speak of velocities as *relative* only when they are relative to a "moving" part of the turbine. To illustrate what is meant, let X (Fig. 74) be part of a turbine wheel moving with an absolute velocity, W, as shown by the arrow. Let V be the absolute velocity of a jet of fluid. Then the velocity of the fluid relatively to the turbine will be obtained by making QB = W, and completing the

parallelogram APBQ, when PB will represent the velocity of the jet relatively to the wheel. This relative velocity PB is the velocity which the jet would have if a velocity were imparted to both the wheel and the jet of an amount sufficient to render the net velocity of the wheel equal to zero. Now, a velocity which would render the net velocity of the wheel

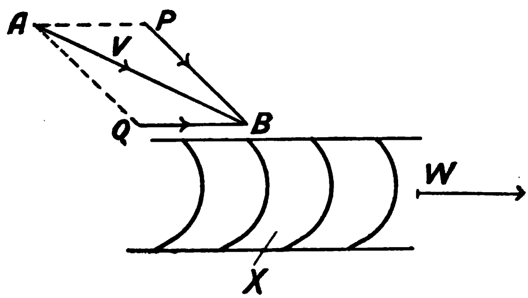


FIG. 74.

equal to zero would be equal and opposite to W. Therefore, combine this velocity with V, and the velocity PB is obtained. Or we may define the velocity of the jet relatively to the wheel as that velocity which, combined with the velocity of the wheel, produces the absolute velocity of the jet. Now, PB represents the velocity which, combined with the velocity of the wheel, produces the absolute velocity represented by AB. Therefore, PB represents the velocity of the jet relatively to the wheel.

In Fig. 75, let V = the absolute velocity of the fluid impinging on the blades or vanes of a turbine; let W = the velocity of the turbine vanes. Then R, the velocity of the fluid relatively to the turbine, can easily be determined. If the course of the fluid is not to be abruptly altered, it is necessary that the vanes where the fluid enters should be parallel to the line of R, and this is usually the case where possible. If the

sectional area of the stream or jet of fluid between two vanes is maintained constant and the volume of the fluid remains constant, then the velocity of the fluid relatively to the vanes will also be constant in magnitude, neglecting friction. Let r represent the velocity of the fluid, leaving the ring of blades relatively to the blades. Then $r = R$ in magnitude: the direction only is altered. A'BCDEF represents the path of the fluid relatively to the blades. That, however, is not the actual or absolute path of the fluid, for the blades themselves have a velocity equal to W. If we combine the velocity W with the relative velocity of the fluid at any point, we get the absolute velocity. Thus at C the absolute velocity of the fluid is represented by GC, at D the absolute velocity of the fluid is represented by HD, and at E the absolute velocity of the fluid is represented by EK. The actual or absolute velocity of the fluid will be in the line ABJLM. EL is the distance through which the blades move while the fluid is moving between the blades from B to E.

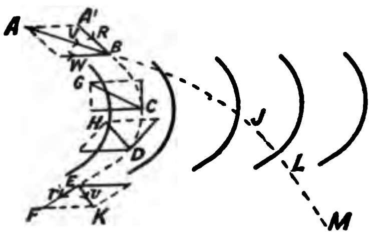


Fig. 75.

The absolute velocity of the fluid when enclosed by the vanes is not important, but the absolute velocities when entering and leaving the rings of vanes are important, as the kinetic energies of the fluid when entering and leaving the rings of vanes are proportional to the squares of these velocities. Let v be the absolute velocity of the fluid when leaving the ring of vanes. Then the kinetic energy given up by the fluid to the turbine will be proportional to $V^2 - v^2$ and the efficiency, neglecting frictional losses, will be $\frac{V^2 - v^2}{V^2}$.

It will be seen that the angle of the vanes, except at the points of entrance and exit, cannot effect the efficiency except through increasing or diminishing the frictional losses. By forming the vanes with a smooth gradual curve, and with the tangent of each at the point of entrance parallel to the relative velocity of the fluid at that point, the frictional velocities may in most cases in an hydraulic turbine be reduced to an almost inappreciable amount. The question of friction in a steam turbine is more difficult.

It is obvious that it will be desirable to have v as small as possible. Now, with a given velocity V, the smallness of v depends upon the velocity W of the vanes, and on the angles of the vanes at the points of entrance and exit of the fluid.

In Fig. 76 let ab represent V in magnitude and direction,

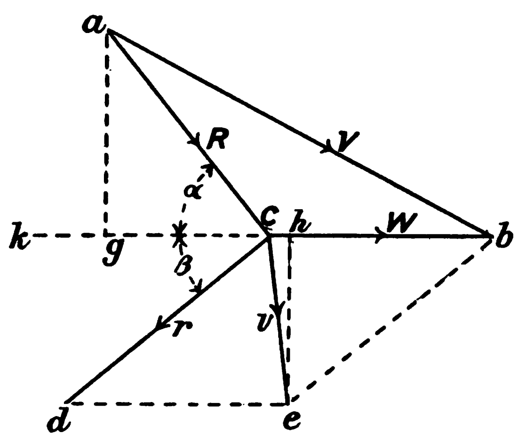


FIG. 76.

and cb represent W in magnitude and direction: then ac represents R in magnitude and direction. Let cd represent r in direction; then, if cd equals ac, cd will also (neglecting friction) represent the magnitude of r. If bc be produced to k, the angle

ack, or α, will represent the angle of the vanes at the point of entrance, and the angle at *dck*, or β, will represent the angle of the vanes at the point of exit. By completing the parallelogram *cbed*, we obtain *ce*, which represents v in magnitude and direction.

Draw *ag* and *eh* perpendicular to *bk*.

$$\text{Then } ce^2 = cb^2 + eb^2 - 2bc\,.\,bh$$

$$\text{Now } eb = cd = ac$$

$$\begin{aligned}\text{Therefore } ce^2 &= cb^2 + ac^2 - 2bc\,.\,bh\\ &= ab^2 - 2bc\,.\,cg - 2bc\,.\,bh\\ &= ab^2 - 2bc(cg + bh)\\ &= ab^2 - 2bc(ac\cos\alpha + eb\cos\beta)\\ &= ab^2 - 2bc(ac\cos\alpha + ac\cos\beta)\\ &= ab^2 - 2bc\,.\,ac(\cos\alpha + \cos\beta) \quad . \quad . \quad . \ (1)\end{aligned}$$

$$\text{Therefore } v^2 = V^2 - 2bc\,.\,ac(\cos\alpha + \cos\beta) \quad . \quad . \quad . \ (2)$$

It is therefore evident that with a given initial absolute velocity of the fluid, v^2 will be the smallest when $2bc\,.\,ac(\cos\alpha+\cos\beta)$ is greatest. It can be seen that this will occur when α and β are each equal to zero, and when $bc = cg$, which in this case will equal *ac*. W would then equal $\frac{1}{2}$V, and the vanes would be as shown in Fig. 77.

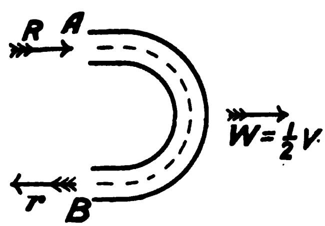


FIG. 77.

If V = velocity at A, the velocity W of the vane should be $\frac{1}{2}$V. The velocity R of the fluid relatively to the vane at A would therefore be $\frac{1}{2}$V. Therefore the velocity r of the fluid relatively to the vane at B would also be $\frac{1}{2}$V, and therefore the absolute velocity of the fluid at this point would be zero.

Not only when the angles α and β are both equal to zero, but in any case when α and β are fixed, and V is also fixed in

magnitude, it can be seen from equation (2) that v^2 is least when $ac \, . \, bc$ (Fig. 76) is a maximum. Since the area of the triangle $abc = \frac{1}{2}ac \, . \, bc \sin a$, and also equals $\frac{1}{2}ab \, . \, cm$ (Fig. 78), where cm is perpendicular to ab, it follows that—

$$\frac{ab \, . \, cm}{\sin a} = ac \, . \, bc$$

Therefore v^2 is least when $\dfrac{ab \, . \, cm}{\sin a}$ is a maximum.

But ab and $\sin a$ are both constant.

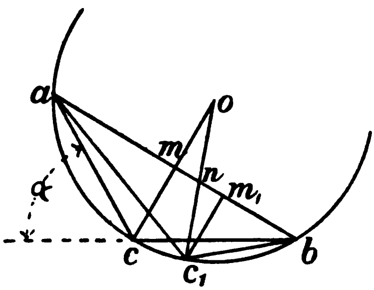


FIG. 78.

Therefore v^2 is least when cm is a maximum.

This occurs when m is the middle point of ab.

For, draw any other triangle, abc' (Fig. 78), on base ab and with angle $ac'b$ = angle acb.

Then the points a, c, c', b are on the circumference of a circle whose centre will be on cm produced.

Let o be the centre.

Join oc', cutting ab at n.

Draw $c'm'$ perpendicular to ab.

Then om is less than on.

Therefore $oc - om$ is greater than $oc' - on$.

Therefore cm is greater than $c'n$, and therefore greater than $c'm'$.

Therefore cm is a maximum when m is the middle point of ab.

Therefore v^2 is least when m is the middle point of ab; that is, when $bc = ac$, or when $bc = \dfrac{ab}{2 \cos \frac{a}{2}}$.

It follows, therefore, that the best value for W is never less than $\frac{1}{2}$V. When α is large, the best value for W is considerably more than $\frac{1}{2}$V.

It is usually impracticable to make α and β equal to zero, although this is very nearly obtained in the Pelton water-wheel. But even when α and β are each equal to 45°, the energy lost need not exceed about 17 per cent. of the whole; for, when $ac = bc$ (Fig. 76), the angle $abc = \frac{1}{2}$ of angle $acg = 22\frac{1}{2}°$; and the angle $ecb = \frac{1}{2}$ of angle $dcb = 67\frac{1}{2}°$.

$$\text{Therefore } ec = \frac{eh}{\sin 67\frac{1}{2}°} = \frac{ag}{\sin 67\frac{1}{2}°} = \frac{ab \sin 22\frac{1}{2}°}{\sin 67\frac{1}{2}°}$$

$$\text{Therefore } ec^2 = \frac{ab^2 \sin^2 22\frac{1}{2}°}{\sin^2 67\frac{1}{2}°} = 0{\cdot}17ab^2$$

With hydraulic turbines V is comparatively small; 100 feet per second is a high value. In steam turbines, however, V is immensely greater.

If steam at a high pressure is allowed to escape through a small, sharp-edged orifice in a plate into the open air or into a chamber at a lower pressure, it is found that only a small portion of its heat is converted into kinetic energy. If, however, the steam is allowed to escape through a diverging nozzle, a much larger proportion of its heat energy is converted into kinetic energy.

Suppose that a pound of dry saturated steam at 300 lbs. pressure absolute is expanded through a divergent nozzle to a chamber communicating with a condenser, and suppose that 20 per cent. of the total heat energy in the steam is converted into kinetic energy; then, as the total heat of the steam equals about 1700 thermal units—

$$\text{K.E.} = 1700 \times \frac{20}{100} \text{ thermal units} = \frac{1700 \times 20}{100} \times 778 \text{ foot-lbs.}$$

$$\text{Therefore } \frac{v^2}{2g} = 1700 \times \frac{20}{100} \times 778$$

$$\text{Therefore } v^2 = \frac{1700 \times 20 \times 778 \times 2g}{100}$$

$$\text{Therefore } v = 4116 \text{ feet per second}$$

If this steam be allowed to act on a single ring of vanes in a steam turbine, then, as we saw that for the greatest efficiency the velocity of the vanes must never, in any case, be less than half the velocity of the entering fluid, it follows that the velocity of the vanes should not be less than 2058 feet per second.

Now, it can be proved that if a ring, whose thickness measured radially is not great compared with its mean diameter, be rotated about its axis, the stress produced in the material due to centrifugal force will be approximately wv^2;* where w

* Let the velocity at outer circumference of the ring shown in Fig. 78A be represented by V_1, and the radius to the outer circumference by R_1.

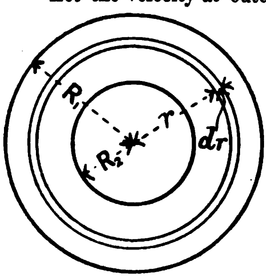


FIG. 78A.

$$\text{Therefore velocity at any radius } r = \frac{V_1 \times r}{R_1}.$$

Let w = density of material.

$$\left.\begin{array}{l}\text{Therefore centrifugal}\\ \text{force at radius } r \text{ on}\\ \text{ring of indefinitely}\\ \text{small breadth } dr\end{array}\right\} = \frac{2\pi r \, . \, dr \, . \, w}{r} \times \frac{V_1^2 \times r^2}{R_1^2}$$

$$= \frac{2\pi \, . \, dr \, . \, w \, . \, V_1^2 \, . \, r^2}{R_1^2}$$

$$\text{Therefore total C.F.} = \int_{R_2}^{R_1} \frac{2\pi dr \, . \, w \, . \, V_1^2 \, . \, r^2}{R_1^2}$$

(where R_2 = interior radius of ring A)

$$= \frac{2\pi w V_1^2}{R_1^2} \int_{R_2}^{R_1} r^2 \, . \, dr$$

$$= \frac{2\pi w V_1^2}{R_1^2} \times \frac{R_1^3 - R_2^3}{3} = \frac{2\pi w V_1^2 (R_1^3 - R_2^3)}{3R_1^2}$$

Let v = velocity at mean radius from centre.

$$\text{Then } v = \frac{R_1 + R_2}{2} \times \frac{V_1}{R_1} \text{ or } V_1 = \frac{2vR_1}{R_1 + R_2}$$

is the density of the material (*i.e.* mass per unit volume), and v is the mean velocity, that is, the velocity at a point at the end of a mean radius. The width of the ring, measured parallel to its axis, and the mean radius do not affect the result. Even if the thickness of the ring, measured radially, is great compared with the mean diameter, the result is not greatly altered. If a steel ring, therefore, weighing 500 lbs. per cubic foot, could have a mean velocity of 2000 feet per second, the stress produced in it would be nearly 200 tons per square inch of cross-section. If the interior diameter of the ring and its velocity there be fixed, then any increase in the external diameter will increase the stress.

This shows that when high-pressure steam is expanded all at one step, the efficiency of a turbine using it is limited by the strength and weight of the materials available for its construction. This difficulty may be overcome by expanding the steam in steps, so that an efficient velocity may be obtained within the limits allowed by the material. On p. 63 it was shown that the velocity of the wheel had to be half, or more than half, of the velocity of the entering fluid, if the velocity of the fluid when leaving the wheel was to be a minimum. But the latter velocity need not be a minimum if the fluid has to act on another series of vanes. The fluid

$$\text{Therefore total C.F.} = \frac{2\pi w(R_1{}^3 - R_2{}^3)}{3R_1{}^2} \times \frac{4v^2R_1{}^2}{(R_1 + R_2)^2} = \frac{8\pi wv^2(R_1{}^3 - R_2{}^3)}{3(R_1 + R_2)^2}$$

$$\text{Now, the force tending to break the ring across a diameter} = \frac{\text{C.F.}}{\pi}$$

$$\text{Therefore average stress of material} = \frac{\text{C.F.}}{\pi \times 2(R_1 - R_2)} = \frac{8\pi wv^2(R_1{}^3 - R_2{}^3)}{3(R_1 + R_2)^2 \times 2\pi(R_1 - R_2)}$$

$$= \frac{4wv^2(R_1{}^2 + R_1R_2 + R_2{}^2)}{3(R_1 + R_2)^2}$$

When R_2 approaches R_1, the stress approaches $\dfrac{4wv^2 \times 3R_1{}^2}{3 \times 4R_1{}^2}$, which $= wv^2$.

may act on several sets of vanes in succession, and the angle and velocities of these vanes may be so arranged that the fluid gives up a portion of its energy to each.

In Fig. 79 let *ab* represent the absolute velocity of the fluid entering the first series of vanes. Let α and β be the angles of the vanes at the points of entrance and exit of the fluid, and let *cb* represent the velocity of the vanes. Then *ac* represents the velocity of the fluid relatively to the vanes as it enters, and *cd* its velocity relatively to the vanes as it leaves. If the sectional area of the fluid while passing through between the vanes is constant, and if the fluid neither expands nor contracts in volume, then $cd = ac$; *ce* will represent the absolute velocity of the fluid as it leaves the first series of vanes. If the fluid be then guided so that it takes the direction *ef*, and if *ef* be made equal in length to *ce*, then *ef* will represent the absolute velocity of the fluid as it enters the second series of vanes. If these vanes are similar to the last, and have the same velocity which is here represented by *mf*, the relative velocity of the fluid entering the vanes will be represented by *em*, and the fluid leaving this series of vanes will have a relative velocity represented by *mn*, which is

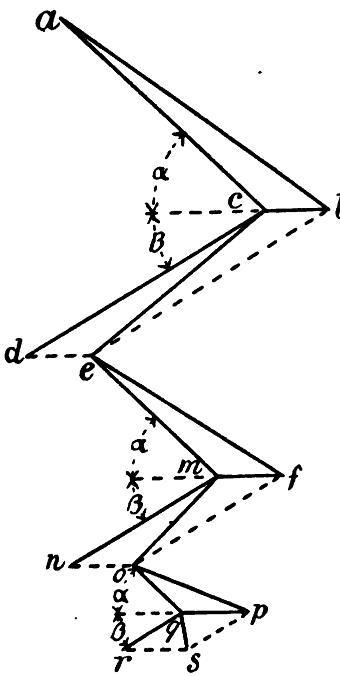


FIG. 79.—Diagram showing Velocities of Fluid in a Compound Turbine, the volume of fluid being constant or increasing proportionately to increase of section of passages.

equal to *em*, and an absolute velocity represented by *mo*. If the fluid be now guided into the direction *op*, and made to act on another series of similar vanes, having a similar velocity represented by *qp*, the fluid will leave this series of vanes with an absolute velocity represented by *qs*. It will thus be seen that the energy taken from the fluid, which is proportional to $ab^2 - qs^2$, is a large proportion of the total available energy, which is proportional to ab^2; but the velocity of the vanes is only a small fraction of the initial velocity of the fluid. By having a greater number of series of vanes, the velocity of these could be kept still lower.

The several series of vanes can be all arranged on the same shaft. If all the series are placed the same distance from the axis of the shaft, *cb*, *mf*, and *qp* will be equal. Otherwise these lines will be unequal.

We have assumed that the fluid neither expands nor contracts in volume from the time it enters the first series of vanes to the time it leaves the last series. If the fluid is a gas, however, it usually will expand during the interval. If the area of section of the passages for the fluid through between the vanes be correspondingly increased, the diagram shown in Fig. 79 will be unaltered. Otherwise the diagram will be modified. Fig. 80 shows a diagram for the same vanes as shown in Fig. 79, and with these vanes having the same velocity, but with the fluid expanding both while passing

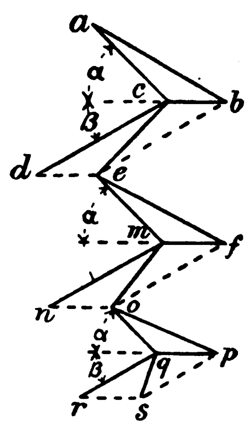


FIG. 80.—Diagram showing Velocities of Fluid in a Compound Turbine, the volume of fluid increasing at a greater rate than section of passages.

through each series of vanes, and in passing from one series to the next. *cd* is, therefore, greater than *ac*, as a greater volume of fluid leaves the first series of vanes than enters it. Similarly, *ef* is greater than *ce*, as a greater volume of fluid enters the second series of vanes than leaves the first series. Similarly, *mn* is greater than *em*, *op* than *mo*, and *qr* than *oq*. The energy taken from the fluid is in this case not proportional to $ab^2 - qs^2$, as some of the initial energy of the fluid exists as heat energy, and is converted into kinetic energy during the passage of the fluid through the apparatus.

In a Parsons steam turbine, practically the whole of the expansion of the steam takes place after the fluid has entered the first series of vanes, and, as the steam passes through a great many series of vanes, its velocity is never excessive. As, moreover, with a number of series of vanes, the velocity of the vanes need only be a small fraction of the velocity of the steam, it follows that vane-speeds can be kept comparatively low without losing the efficiency. Very good results have been obtained with Parsons turbines running at nearly as low a speed as that of fast reciprocating engines.

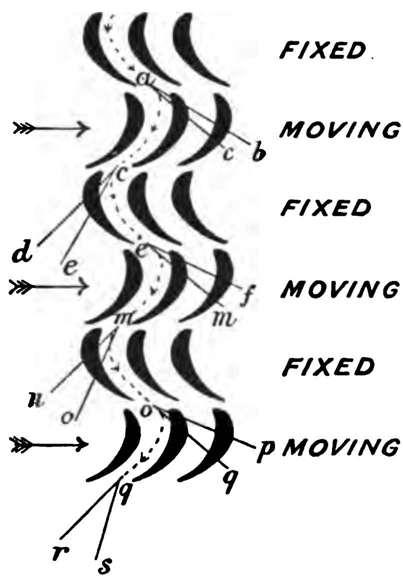


FIG. 81.—Passage of Steam through a Parsons Turbine.

Fig. 81 shows the fixed and moving vanes or blades of a parallel-flow turbine of the Parsons type, the dotted line and

small arrow-heads showing the passage of the steam. The fixed blades are for guiding the steam from one series of moving blades to the next. The relative and absolute velocities at different points are lettered to correspond with Fig. 80. The lines *ab*, *ac*, etc., are intended to represent only the directions, and not the magnitudes, of the velocities.

In order to reduce the velocity of the fluid acting on the vanes of a steam turbine, it has been proposed to cause the steam-jet, by an injector action, to draw in air, water, or other fluid at atmospheric pressure, the velocity of the combined fluid being thus made moderate. This would allow of a lower efficient vane speed; but, as a greater mass of fluid would leave the turbine, and as this fluid must have a certain velocity, the energy thus lost would be increased without any increase in the energy entering the turbine.

CHAPTER VI.

ENTROPY AND ENTROPY-TEMPERATURE DIAGRAMS.

As we shall be dealing with entropy-temperature diagrams, and as this subject is not very well known, it may be advisable, in the first place, to explain what is meant by "entropy," and what can be determined by an entropy-temperature, or, as it is sometimes called, a theta-phi diagram. To an engineer accustomed only to diagrams in which the ordinates and abscissæ represent readily appreciable quantities, such as pressure, or volume, or steam consumption, the idea of entropy is rather difficult to grasp. This "ghostly quantity," as Professor Perry calls it, is not perceptible by the senses, and cannot be measured directly by any gauge or meter. It is, nevertheless, a very convenient term of expression, and entropy-temperature diagrams are very instructive and very useful.

In an ordinary pressure-volume or pressure-distance diagram, as, for example, an indicator diagram, the ordinates represent pressure, the abscissæ represent volume or distance travelled, and the areas represent energy received or rejected, or work done. Now, when heat is put into or taken out of a substance, any small part of the heat so dealt with is equal to the temperature at which it was put in, or taken out, multiplied by some quantity. This quantity is called change of entropy, or difference of entropy.

In order that the temperature may be constant while the small amount of heat is being put in or taken out, it is necessary for a general case that the small part of the heat should be indefinitely small. Suppose this to be the case.

Let ϕ represent entropy, and $d\phi$ an indefinitely small change of entropy.

Let Q represent quantity of heat put into or taken from a substance, and dQ an indefinitely small change in the quantity of heat held by the substance.

Let τ represent temperature, and $d\tau$ an indefinitely small difference of temperature.

Then by our definition $dQ = \tau \times d\phi$.

Now, the total heat supplied to or taken from the body equals the sum of all the indefinitely small parts, and therefore equals the sum of all the items $\tau \times d\phi$.

This is expressed by saying

$$Q = \int dQ = \int \tau d\phi$$

This may be expressed in the form—

$$\phi_2 - \phi_1 = \int_{\tau_1}^{\tau_2} \frac{dQ}{\tau} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

If, therefore, we draw a curve, ACB (Fig. 82), such that its ordinates A*a*,C*c*,B*b* (that is, the distances of points in it from OX) represent temperature, and such that the areas *a*AC*c* and *c*CB*b* enclosed between the curve, the line OX, and any two ordinates, represent quantities of heat put into or taken out of a substance, then it is clear that the distances *ac* and *cb* will represent differences of entropy, and that the abscissæ O*a*, O*c*, O*b* (or distances of points in the curve from OY) will represent the entropy of the substance at the points A, C, B.

Temperatures are reckoned from absolute zero, which is represented by the line OX. Entropy may be reckoned from any point, but it is convenient, in dealing with water and steam, to consider zero entropy to be that of water at freezing-point (32° F.). This will then be represented by OY. Quantities of heat always refer to one pound of the working substance.

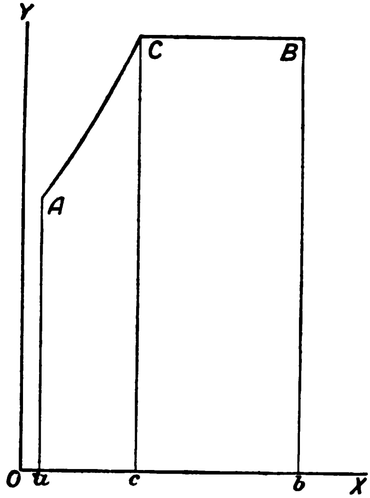


FIG. 82.—Entropy-temperature Diagram.

In Fig. 83 AB is an entropy-temperature curve for water raised in temperature from 100° F. to 400° F. The temperatures are indicated both on the ordinary Fahrenheit and on the absolute scale. It will be seen that the difference of entropy between water at 100° F. and water at 400° F. equals 0·437. The amount of heat required to effect this physical change in the water is represented by the area $aABb$. The curve AB

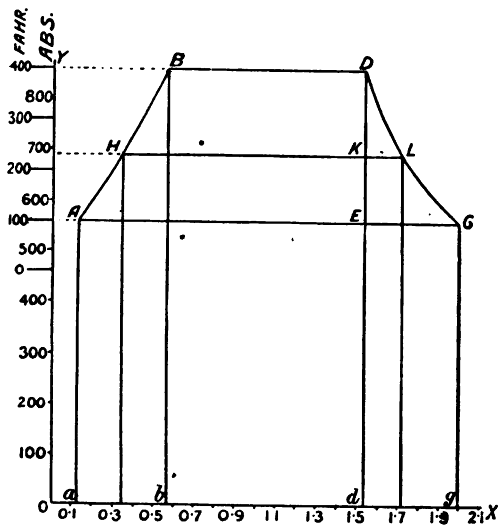


FIG. 83.—Entropy-temperature Diagram for Water and Steam.

may be drawn by obtaining from a table the entropy of water at several temperatures, and plotting these values. If the water at 400° F. be converted into steam at that temperature, the entropy-temperature curve will be parallel to OX, as the temperature is unchanged. The change of entropy will be represented by *bd*, and the heat put into the substance by the area BD*db*. The heat put into the water is obviously the latent heat of steam at 400° F. (860° abs.). This equals 830 units. The change of entropy *bd* is therefore equal to $\frac{830}{860}$, or 0·965, as indicated on the diagram. If, now, the steam expand adiabatically against a resistance, the temperature will fall, but as no heat is being imparted to or taken from the steam, it is obvious that the area below the curve of expansion must be zero—that is, that no change of entropy will take place. The entropy-temperature curve will therefore lie along D*d*, and will be represented by DE if the temperature fall to 100° F. If heat be now abstracted from the steam and water (for some of the steam will have condensed during expansion) till all the fluid exists in the liquid state, but without lowering the temperature, the entropy-temperature curve will be EA, which is parallel to OX. The quantity of heat taken from the fluid will be represented by the area *a*AE*d*. The total heat supplied to the fluid is therefore proportional to the area *a*ABD*d*, and the heat abstracted to the area *a*AE*d*. The heat converted into work is therefore proportional to the area ABDE, and the efficiency of a heat-engine working on this cycle $= \frac{\text{area ABDE}}{\text{area } a\text{ABD}d}$.

If, instead of allowing the steam to expand adiabatically, we had, during expansion, supplied just sufficient heat to it to maintain it in a dry, saturated condition, the entropy-temperature curve for expansion would be DG instead of DE. If heat had

then been taken from the steam without reducing its temperature till the whole had condensed, the drop of entropy would be represented by AG (or *ag*), and the quantity of heat abstracted by the area *a*AG*g*. This last quantity is obviously the latent heat of steam at 100° F., and it is evident that the ratio of the area *a*AE*d* to the area *a*AG*g* is the fraction of the latent heat available to be given up after the steam has expanded, according to the line DE. This ratio must therefore represent the amount of steam uncondensed at E. The areas are proportional to the lines AE and AG, and therefore $\frac{AE}{AG}$ represents the dryness fraction, or the fraction of the steam uncondensed after the adiabatic expansion DE has taken place, or when the point E is reached during the isothermal withdrawal of heat GA. Similarly, if any other horizontal line such as HKL be drawn, $\frac{HK}{HL}$ will represent the dryness fraction of the steam at the point K of the adiabatic expansion DE.

The curve DG may be drawn by obtaining from a table the entropy of dry, saturated steam at several temperatures, or it may be obtained in another manner. $AG \times Aa =$ area *a*AG*g*. But *a*A represents a certain temperature, and *a*AG*g* represents the latent heat of steam at that temperature. Therefore the length of AG can be obtained by dividing the latent heat by the temperature. Several horizontal lines, such as AG and HL, can thus be determined, and the curve DLG drawn through their ends.

CHAPTER VII.

THEORETICAL CONSIDERATION OF DIFFERENT TREATMENTS OF STEAM IN A HEAT-ENGINE.

It is intended in this chapter to consider the effects of treating steam in different ways on the efficiencies of heat-engines with special reference to the steam turbine.

Let us consider the transfer of heat energy into mechanical energy in a heat-engine or apparatus comprising a boiler in which water is heated to a certain temperature and then converted into steam, a turbine or other motor in which the steam is expanded and loses some of its heat, and a condenser in which more of the heat is taken from the fluid before the latter is returned to the boiler.

The different cases which will be considered have been chosen not to represent what occurs in practice, but to indicate the effects of different treatments of the steam, so that it can be ascertained what had best be done with any type of turbine, in order to prevent waste and promote efficiency, and what is likely to be gained or lost by any alteration in treatment, such, for example, as by superheating the steam.

Case I.

Let us suppose, in the first instance, that feed water is received into a boiler at 85° F., and heated to 382° F. The

entropy-temperature curve for this heating is shown at AB, in Fig. 84.

Suppose that the water is converted into steam at this temperature, which means that the pressure is 200 lbs. absolute. This absorption of heat is represented by BC. Let the steam now expand adiabatically, doing work, till the pressure falls to 0·6 lb. absolute. The temperature corresponding to this pressure is 85° F. This expansion is represented by the line CD on the diagram. Some of the steam will condense during this expansion, and we can find the wetness at any point in CD, by the method described in connection with Fig. 83. Lastly, let heat be abstracted from the fluid till the whole of the steam has condensed, but without any reduction of temperature, and let the water be returned to the boiler. This action is represented by DA on the diagram (Fig. 84). It does not matter whether the heat be abstracted from the steam in the turbine or in a condenser, or in any other vessel, provided that it takes place after the expansion and the fall in temperature are completed.

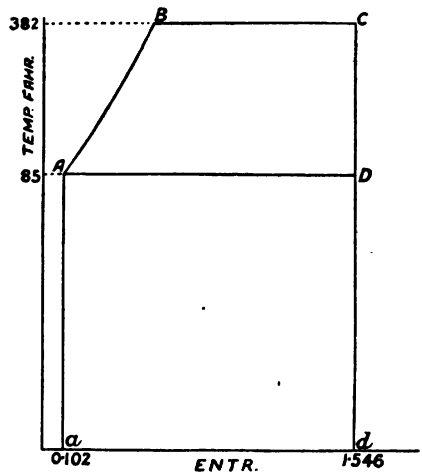


Fig. 84.—Case I.: Adiabatic Expansion; isothermal compression; range of temperature, 85° F.—382° F.

The heat supplied to the fluid is then represented by the area aABCd, and the heat abstracted by the area aADd. The heat converted into work is therefore represented by the area ABCD and—

$$\text{The efficiency} = \frac{\text{area ABCD}}{\text{area } a\text{ABC}d} = 0{\cdot}31$$

Case II.

Let us suppose now that the steam generated at 200 lbs. pressure, instead of expanding adiabatically, be supplied during expansion with sufficient heat to prevent any condensation. This might be approximately attained by jacketing a steam turbine with high-temperature steam. The condensation will then all take place at constant temperature, as shown by EA. The entropy - temperature diagram will then be as shown in Fig. 85.

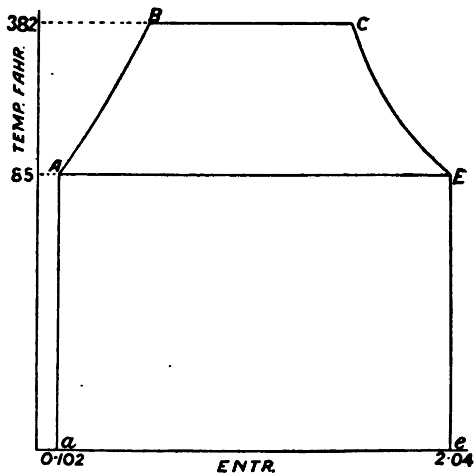


Fig. 85.—Case II.: Expansion along Line of Dry Saturated Steam; isothermal compression; range of temperature, 85° F.—382° F.

The heat supplied to the fluid is represented by the area aABCEe, and the heat abstracted by the area aAEe. The heat converted into work is therefore represented by the area ABCE, and—

$$\text{The efficiency} = \frac{\text{area ABCE}}{\text{area } a\text{ABCE}e} = 0{\cdot}28$$

Compared with Case I. it will be seen that there is an increase both in the heat supplied and in that converted into work, but the latter is not increased proportionately to the former, and hence the drop in the efficiency.

Case III.

Suppose in this case that the steam, instead of receiving heat while expanding, has heat taken from it, as, for example, by radiation from the exterior of a turbine. Suppose that by this means twice as much steam is condensed during expansion as in Case I. The entropy-temperature diagram will then be as shown in Fig. 86, where $\frac{FE}{AE}$ is the fraction of the steam that is condensed at the end of the expansion. The remainder of the steam is condensed at constant temperature, as shown by the line FA. The heat supplied to the fluid is represented by the area aABCd, and the heat withdrawn by the area aAFCd. The heat converted into work is therefore represented by the area ABCF, and—

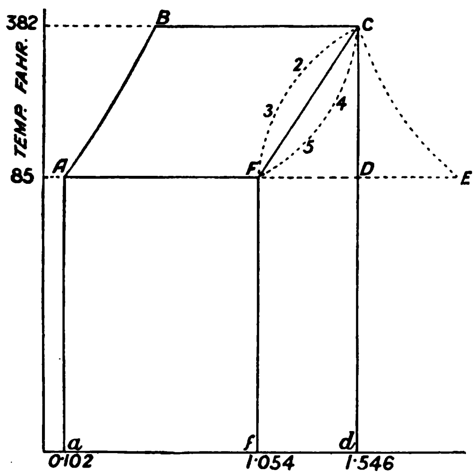


Fig. 86.—Case III.: Expansion with Leakage of Heat; isothermal compression; range of temperature, 85°—382° F.

$$\text{The efficiency} = \frac{\text{area ABCF}}{\text{area } a\text{ABC}d} = 0{\cdot}25$$

The heat supplied is the same as in Case I., but the portion of this that is converted into work is less than in Case I. by the amount represented by the triangle CFD. The efficiency is therefore less than in Case I.

The total leakage of heat during expansion is represented

by the area fFCDd. Of this the area CFD goes to reduce the heat converted into work, and the area fFDd to reduce the work of the condenser. There is no reason why CF should be a straight line. A straight line has only been assumed for convenience. The nature of the line would depend on the construction of the engine. (In a reciprocating engine the line is usually very convex to the left, on account of the coolness of the cylinder at the beginning of the stroke.) It should therefore be noted that if the leakage of heat had been greater than that represented towards the beginning of the expansion, and less towards the end, so that the line of expansion was as shown by the dotted line C23F, the efficiency would have been reduced; while if, on the contrary, the leakage had been less towards the beginning of the expansion and greater towards the end, as indicated by the dotted line C45F, the efficiency would have been increased; the total amount of steam condensed being the same in all cases.

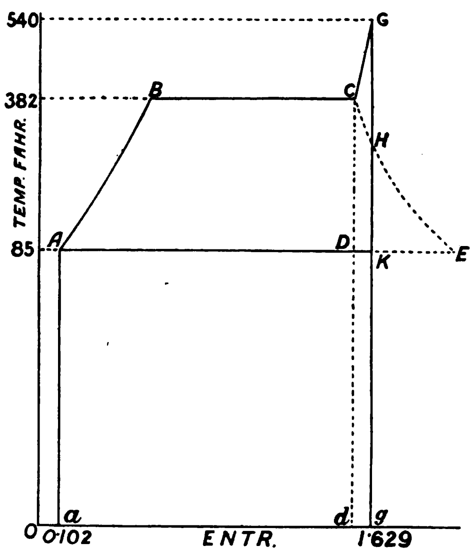


FIG. 87.—Case IV.: Superheating; Adiabatic Expansion; isothermal compression; range of temperature, 85° F.—540° F.

CASE IV.

Suppose in this case the steam is superheated before expansion takes place, from 382° F. to 540°, and that the expansion is then adiabatic. Fig. 87 is the diagram for this case,

CG being the curve for the superheating action. It will be seen that the line GK of adiabatic expansion cuts the line CE of dry, saturated steam at the point H. This indicates to us that, during the fraction of the adiabatic expansion represented by GH, the steam is superheated; while, during the remaining fraction represented by HK, it is wet—at the point H it is dry and saturated. The heat supplied to the fluid is represented by the area aABCGg, and the heat abstracted by the area aAKg. The heat converted into work is therefore represented by the area ABCGK, and—

$$\text{The efficiency} = \frac{\text{area ABCGK}}{\text{area } a\text{ABCG}g} = 0{\cdot}32$$

The heat supplied to the fluid during the superheating action is represented by the area dCGg. Of this the portion represented by the area DCGK is converted into work. The fraction of the heat supplied which is converted into work is therefore greater during this action than during the actions of heating the feed water and generating the steam, and it is this which raises the efficiency slightly above that in Case I.

To draw the curve CG we must make an assumption regarding the specific heat of steam at constant pressure. Let this specific heat be denoted by K, and let us assume that K is a constant, and equal to 0·48. Then from equation (3), p. 71—

$$\phi_2 - \phi_1 = \int_{842}^{1000} \frac{dQ}{\tau}$$

the numbers 1000 and 842 denoting the temperature on the absolute scale, and ϕ_1 and ϕ_2, denoting the entropy respectively before and after the superheating action.

Now $dQ = \mathrm{K}d\tau$.

$$\text{Therefore } \phi_2 - \phi_1 = \int_{842}^{1000} \frac{K d\tau}{\tau} = 0{\cdot}48 \int_{842}^{1000} \frac{d\tau}{\tau}$$

$$= 0{\cdot}48(\log_e 1000 - \log_e 842) = 0{\cdot}48 \times 0{\cdot}1720$$

$$= 0{\cdot}08256.$$

This is the difference of entropy between C and G, and determines the length *dg*. The height *g*G is of course determined by the temperature, namely, 540°. Any other point on the curve CG can be similarly located, and the curve thus obtained.

Case IVa.

In the case just described the higher limit of temperature and the range of temperature exceed that in the other cases, and therefore, in order to make a fair comparison, we must consider the case of an engine working on a cycle, as in Case IV., but with the same limits of temperature as in Cases I., II., and III.

Let us suppose, then, that steam is generated at 224° F., and superheated to 382° F., the cycle otherwise being the same as in Case IV. The entropy-temperature diagram will then be as shown in Fig. 88, where the heat supplied to the fluid is represented by the area

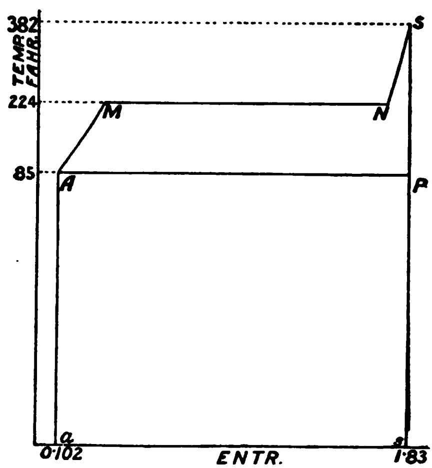


Fig. 88.—Case IVa.: Superheating; Adiabatic Expansion; isothermal compression; range of temperature, 85° F.—382° F.

aAMNSs, and the heat withdrawn by the area aAPs. The heat converted into work is therefore represented by the area AMNSP, and—

$$\text{The efficiency} = \frac{\text{area AMNSP}}{\text{area } a\text{AMNS}s} = 0{\cdot}20$$

The efficiency is less than in Case I., because, although the maximum and minimum temperatures are the same as in Case I., most of the heat is absorbed by the fluid when at a lower temperature.

Case V.

If in Case IV. (Fig. 87), the fluid, instead of expanding adiabatically, had had the same amount of heat abstracted from it during expansion as in Case III. (Fig. 86), the entropy-temperature diagram would be as shown in Fig. 89, where the total leakage of heat is represented by the area qQJGg, which corresponds to and equals the area fFCDd in Fig. 86. The heat supplied, the heat withdrawn, and the heat converted into work are represented respectively by the areas aABCGg, aAQJGg, and ABCGJQ.

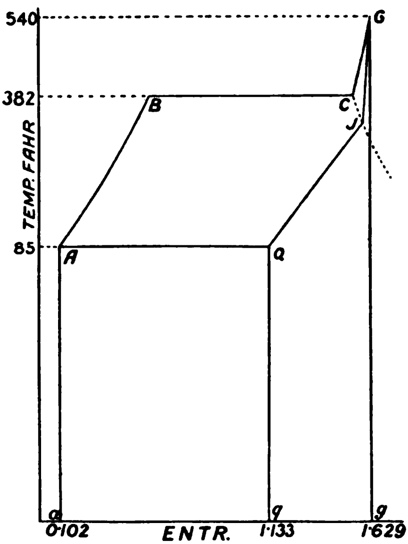


Fig. 89.—Case V.: Superheating; Expansion with Leakage of Heat; isothermal compression; range of temperature, 85° F.—540° F.

$$\text{The efficiency} = \frac{\text{area ABCGJQ}}{\text{area } a\text{ABCG}g} = 0{\cdot}26$$

The increase in efficiency in this case over Case III. is about the same as the increase in Case IV. over Case I., and for the same reason.

Case VI.

In the cases heretofore considered the steam has expanded doing work. But the steam may expand without doing work, or against an imperfect resistance. Joule found by experiment that when a gas expands without doing work, its temperature remains constant. Joule's experiment consisted in placing in a tank of water two vessels, one containing a gas under pressure, and the other empty. On communication being established between the vessels, some of the gas rushed from one vessel to the other, and the pressure fell; but it was found, after equilibrium had been established, that the temperature was the same as at the beginning of the experiment.

The phenomenon of unresisted expansion occurs when steam is passed through a reducing-valve, when the pressure falls and the steam expands without any appreciable amount of work being done. Imperfect resistance to expansion also occurs when steam passes at a high velocity through a restricted opening, and is well known in such a case by the name of "wire-drawing." When unresisted or imperfectly resisted expansion takes place, some of the heat of the gas is converted into kinetic energy; but if the gas has its velocity arrested, the energy returns to the form of heat. Thus, in the case of a reducing-valve, when the valve opens, there is a rush of steam through it, some of the heat energy of the steam being converted into kinetic energy. The rush is, however, arrested at the other side of the valve, and the kinetic energy is returned by impact or eddies to the form of heat.

In Fig. 90 AB represents the heating of the feed water, BC the generation of steam, and CX the adiabatic expansion of the steam, as in the previous cases. Suppose that free expansion then takes place till the steam is completely dried and is superheated. The state of the steam will then be represented by the point Y on the diagram. If the steam had been dried, but not superheated, the point Y would have been on the curve CZ. We cannot connect X and Y by a straight line to represent the free expansion, as this would indicate that heat had been absorbed by the steam, which is not the case. If we want an unbroken curve, we must connect X and Y by means of the straight lines X*d*, *de*, and *e*Y. Y*e* of course equals X*d*, as the temperature is unchanged. Suppose that, after the free expansion, the steam expands adiabatically, doing work, till the temperature falls to 85° F. This expansion is represented in the diagram by YE. The isothermal compression EA completes the diagram, as in the previous cases. The amount of free expansion has been so chosen that, after the last expansion, the steam is dry and saturated, as indicated, by the point E being on the curve CZ.

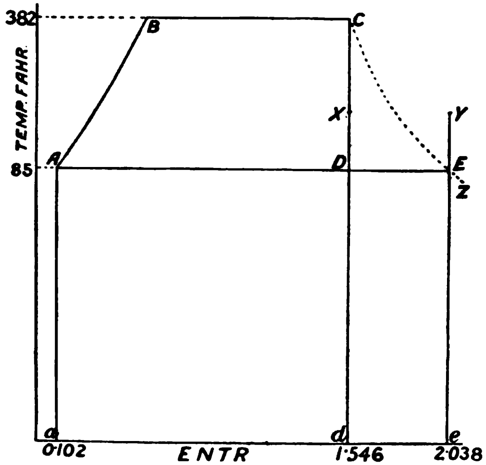


FIG. 90.—Case VI.: Expansion, partly adiabatic and partly unresisted; isothermal compression; range of temperature, 85° F.—382° F.

The heat absorbed by the fluid in this case is represented by the area *a*ABC*d* and the heat rejected by the area *a*AE*e*.

The heat converted into work is represented by the area aABCd—the area aAEe and

$$\text{The efficiency} = \frac{\text{area } a\text{ABC}d - \text{area } a\text{AE}e}{\text{area } a\text{ABC}d} = 0{\cdot}08$$

CASE VII.

In this case let us suppose that the feed water is heated, the steam generated, and the adiabatic expansion commenced as in Case I.; but let the adiabatic expansion continue only till the temperature falls to 250° F., as indicated by the point V. Then let heat be abstracted from the fluid *at constant volume* till temperature 85° F. is reached, as indicated by U, Fig. 91. The isothermal compression UA completes the diagram.

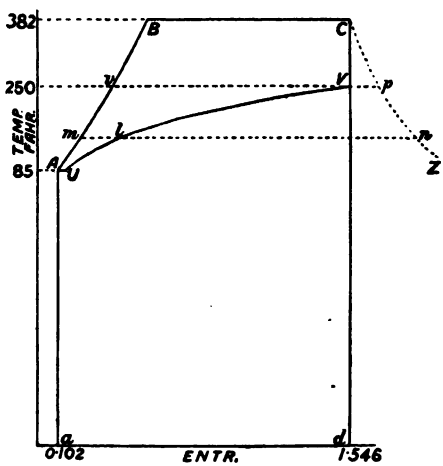


Fig. 91.—Case VII.: Adiabatic Expansion; heat rejected at constant volume, followed by isothermal compression; range of temperature, 85° F.—382° F.

The curve VU is obtained as follows. Let a' be the volume of 1 lb. of saturated steam at 250° F. (the temperature at V), and let a be the volume of the same at any other temperature, τ, between V and U. Let q' be the dryness fraction of the steam at V, and q be the dryness fraction at the temperature τ. Then, neglecting the volume of the water—

$$qa = q'a'$$

because the fluid is expanding at constant volume. If l is

the point on the curve where the temperature is τ, and mln is a horizontal line drawn to meet the line of saturated steam CZ—

$$q = \frac{ml}{mn}$$

$$\text{therefore } ml = q \times mn = \frac{q'a'}{a}mn$$

q' of course equals $\frac{v\mathrm{V}}{vp}$, and a' and a can be obtained from a table of the properties of saturated steam. Hence ml can be obtained. Similarly, any number of other points can be obtained on the curve VU.

The heat supplied to the fluid in this case is represented by the area aABCd, and the heat rejected by the area aAUVd. The heat converted into work is represented by the area ABCVU, and

$$\text{The efficiency} = \frac{\text{area ABCVU}}{\text{area } a\text{ABC}d} = 0{\cdot}18$$

This treatment, by which part of the heat is rejected at constant volume, and part at constant temperature, gives a reduced efficiency compared with the treatment in Case I., where all the heat rejected was given up at constant temperature. In reciprocating condensing engines the heat is commonly rejected, neither on a constant volume line nor on a constant temperature line, but on a line between the two. The nature of the rejection of heat in a steam turbine is pretty much a matter of conjecture.

Case VIII.

Suppose in this case that the feed water is heated, and the steam generated, superheated, and expanded adiabatically, as in Case IV., till the point T (Fig. 92) is reached, where the

temperature is 250° F. Let the fluid now expand at constant volume, as in Case VII., till the point W is reached, when the temperature is the same as at A, and let the cycle be completed by the isothermal compression WA. In this case the heat supplied to the fluid is represented by the area

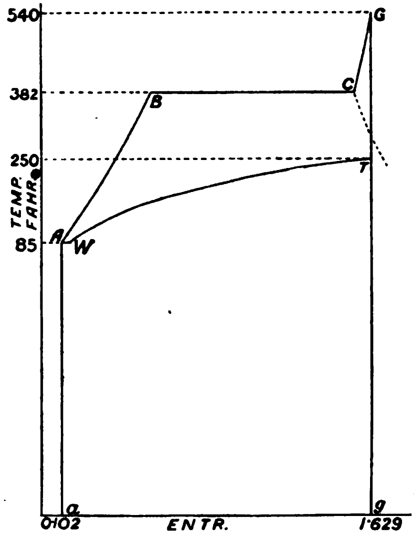


FIG. 92.—Case VIII.: Superheating; Adiabatic Expansion; heat rejected at constant volume, followed by isothermal compression; range of temperature, 85° F.—540° F.

aABCGg, as in Case IV., and the heat rejected is represented by the area aAWTg. The heat converted into work is represented by the area ABCGTW, and

$$\text{The efficiency} = \frac{\text{area ABCGTW}}{\text{area } a\text{ABCG}g} = 0{\cdot}19$$

Many more cases might be studied, but sufficient have been considered to show the effect of different treatments of the steam. The results are here tabulated in Table I.

TABLE I.

Case.	Method of treatment.	Max. temp. F.	Min. temp. F.	Efficiency.
I.	Adiabatic expansion. Isothermal compression	382	85	0·31
II.	Expansion along line of dry saturated steam. Isothermal compression	382	85	0·28
III.	Expansion with leakage of heat. Isothermal compression	382	85	0·25
IV.	Superheating. Adiabatic expansion. Isothermal compression	540	85	0·32
IVA.	Superheating. Adiabatic expansion. Isothermal compression	382	85	0·20
V.	Superheating. Expansion with leakage of heat. Isothermal compression	540	85	0·26
VI.	Expansion, partly adiabatic and partly unresisted. Isothermal compression	382	85	0·08
VII.	Adiabatic expansion. Heat rejected at constant volume, followed by isothermal compression	382	85	0·18
VIII.	Superheating. Adiabatic expansion. Heat rejected at constant volume, followed by isothermal compression	540	85	0·19

It should be borne in mind, however, that a change in the range of temperature will alter the relative efficiencies. It should also be remembered that arbitrary quantities have, as a rule, been chosen for the amount of superheating, amount of free expansion, etc.; and that, if these are altered, the results may be considerably modified. And it must not, above all things, be forgotten that there are practical considerations which affect the efficiency. For example, there is the fluid friction in a turbine. It is probable that the diminution of this fluid friction by superheating the steam accounts in part for the increased economy obtained by superheating; for the results obtained by the tests of Parsons turbines show a greater percentage increase in efficiency with superheating than is due to thermo-dynamic reasons. Table II. shows the effect of superheating on the steam consumption of a Parsons turbine.

TABLE II.

TEST OF 500-KILOWATT TURBO-ALTERNATOR CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO. FOR THE CORPORATION OF BLACKPOOL.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. (Bar. = 30″.)	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw. hr.
146	70	27·1	2500	515	11,000	21·35
150	0	27·0	2500	502	11,600	23·1
135	0	27·3	2500	497	11,953	24·0
133	66	27·3	2500	507	10,693	21·1

CHAPTER VIII.

THE DE LAVAL STEAM TURBINE.

About 1882, Dr. Gustaf de Laval invented a turbine on the principle of Hero's engine. This turbine is illustrated diagrammatically in Figs. 93 and 94. Steam (or other fluid) entered the casing *a* by the nozzle *b*, and passed along the curved hollow arms *c, c*. These arms were formed like the buckets of an outward-flow hydraulic turbine, and the passage of the steam along them caused them to revolve and to rotate the shaft *d*. This shaft drove another shaft at a slower speed by means of friction wheels. The requisite pressure between the surfaces of these wheels was obtained by utilizing the axial thrust of the turbine wheel. The turbine shaft *d* was supported in bearings which allowed it an axial movement. This shaft (see Fig. 95) carried a bevel friction wheel *e*, and the axial thrust of the turbine wheel forced this bevel wheel against the bevel wheel *f* carried by the power shaft *g*.

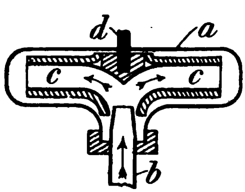


Fig. 93.

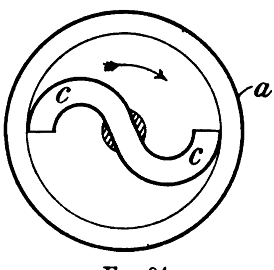


Fig. 94.
Early Turbine of Dr. De Laval's.

In 1889 Dr. De Laval applied for a British patent* for

* No. 7143 of 1889.

a steam turbine wheel combined with a diverging nozzle for the steam supply. The nozzle shown in the specification of this patent was shaped as illustrated in Fig. 96. The steam expands in passing from the smaller section *m* to the larger section *n*, and its velocity increases while its pressure falls. The object is of course to obtain a great kinetic energy with which to act on

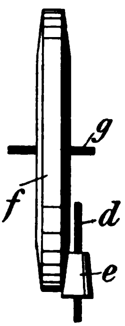


FIG. 95.—Friction Gearing.

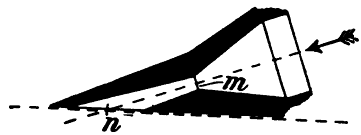


FIG. 96.—De Laval Nozzle.

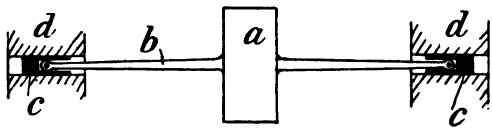


FIG. 97.—Flexible Shaft Support.

the turbine vanes. The manner in which the steam is directed on to the vanes can be seen by referring back to Figs. 2 and 2A.

Another patent of De Laval's of the same year,* refers to the flexible support of steam turbines or other bodies intended to rotate at high velocities. Figs. 97 to 106 illustrate diagrammatically several devices covered by the patent for allowing a certain amount of lateral movement to the rotating mass, to enable it to compensate for slight want of balance.

In Fig. 97, the rotating body *a* is carried on a flexible shaft, *b*, whose ends are placed in the shoes *c*, *c*, which rotate in the bearings *d*, *d*.

In Fig. 98, the rotating body *a* is flexibly connected to the shaft *b*, by providing the latter with a flange, *e*, and inserting

* No. 12,509 of 1889.

rubber rings f, f, as shown. The body is of course also supported by another shaft at the other side.

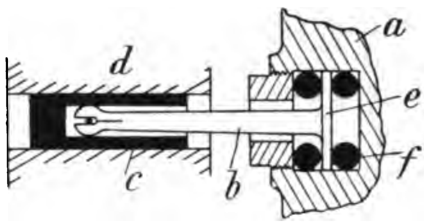


FIG. 98.—Flexibility given by Rubber Rings.

In Fig. 99, spiral springs g, g, are substituted for the rubber rings.

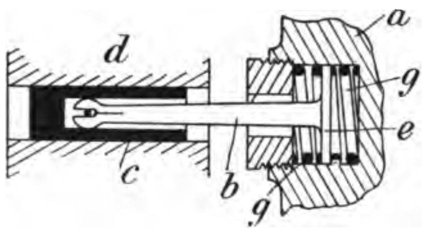


FIG. 99.—Flexibility given by Spring.

In Fig. 100, the shaft b is connected to the rotating body by

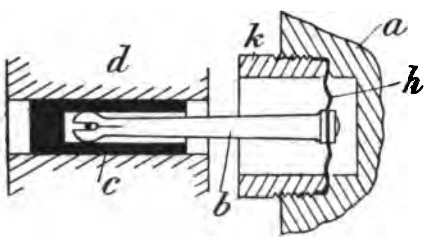


FIG. 100.—Flexibility given by Diaphragm.

means of the flexible diaphragm h, held in place by the gland k.

In the device shown by Figs. 101, 102, 103, in end elevation, side elevation, and section respectively, the shaft b is

supported at each end in bushes *m*, which, by means of the transverse pins *n*, *n*, can swing in the standards *o*.

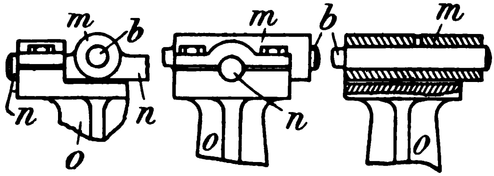


Fig. 101. **Fig. 102.** **Fig. 103.**
Flexibility given by Transverse Pivots.

In Figs. 104 and 105, the bearing bush *p* (one of these is

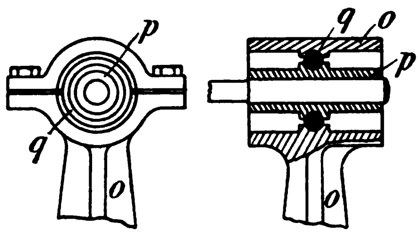


Fig. 104. **Fig. 105.**
Flexibility given by Rubber Ring.

provided at each end of the shaft) is supported in the cylindrical top of the standard *o*, by means of the rubber ring *q*.

In Fig. 106 the shaft is provided with spherical end pieces, *r*.

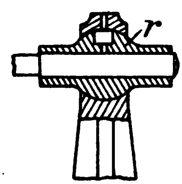


Fig. 106.—Flexibility given by Spherical End Pieces.

British patent, No. 20,603 of 1891, granted to Dr. De Laval, has reference to the exhaust passage from the turbine, which is constructed of a divergent shape in order to produce an ejector action. The velocity of the fluid at the outer end of the nozzle is less than at the inner end, owing to the increase in the section of the passage, and consequently the pressure at the inner end is less than at the

outer end. If, therefore, the pressure at the outer end is atmospheric, a partial vacuum will exist at the inner end of the passage and around the wheel, thus diminishing friction.

Fig. 107 shows a De Laval turbine-dynamo, as constructed

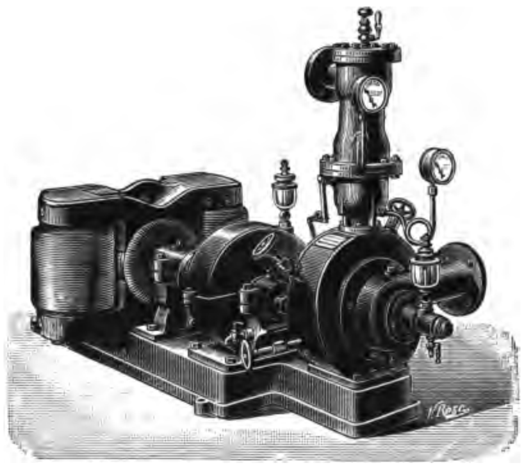


FIG. 107.—De Laval Turbine-dynamo.

by the Société de Laval (France), for horse powers from 5 to 30. The cylinder to the right contains the turbine wheel, and the intermediate cylinder is the gear box in which the high rotary motion of the wheel is geared down to a speed suitable for driving the dynamo, which is shown at the left of the figure.

Fig. 108 shows the principal parts of a turbine such as that shown in Fig. 107, but fitted with a pulley instead of being connected with a dynamo. A is the turbine shaft on which is mounted the disc or wheel B, furnished with a series of vanes. These vanes can also be seen in Fig. 109, where they are lettered W. C is a double helical pinion which gears with the toothed wheel M, the teeth on the wheel and pinion being formed at an angle of 45°, as is shown in the figure. Great

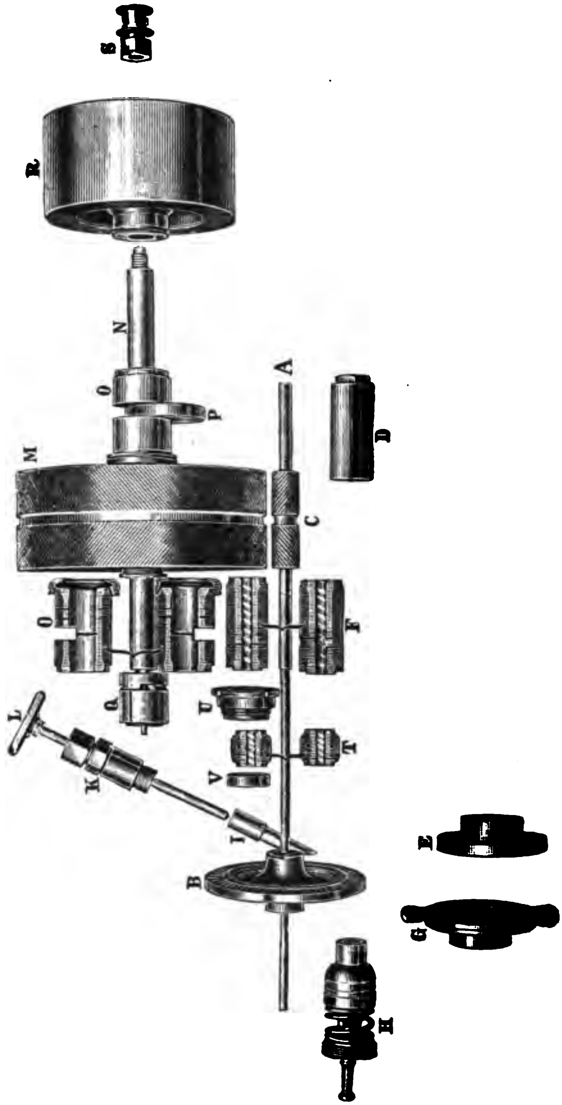


FIG. 108.—Component Parts of De Laval Turbine.

(The block for this has been prepared from an illustration kindly supplied by Messrs. Greenwood and Batley, of Leeds.)

strength is not required in these teeth, as the forces exerted on them are not excessive, owing to the high speed of rotation of the shaft A, and the small diameter of the pinion. It can be seen by a small calculation that, if the diameter of the pinion C be, say, one inch, and the speed of rotation 24,000 revolutions per minute, the force exerted on the teeth will be very small, even if the motor be of considerable power. D is the end bush of the turbine shaft, and F the middle bush, made in

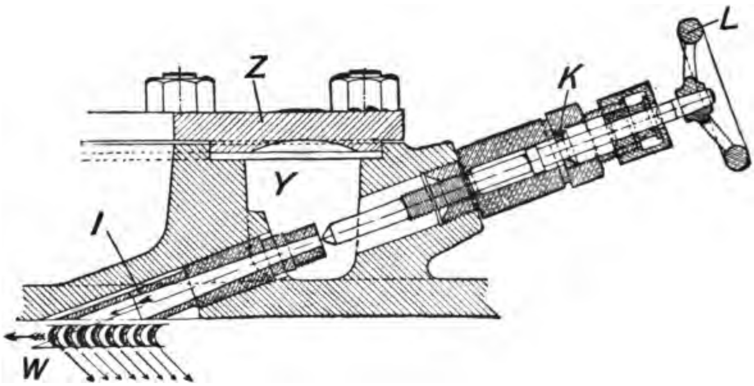


FIG. 109.—Nozzle and Vanes of a De Laval Turbine.

two parts. T is a tightening bush, also made in two parts. O, O, are the gear-wheel shaft bushes which support the power shaft N, which carries the gear wheel M, and the driving pulley R. S is a stop nut for the power shaft, and H a ball bush with adjusting spring for the turbine shaft. U is an adjusting nut, and V a friction gland. I is a steam nozzle, of which several are usually provided, distributed round the wheel. K is the stuffing-box for the spindle stop-valve, which can be actuated by the hand-wheel L. P is a lubricating ring, and Q is the governor which is mounted on the power shaft.

A section of a De Laval governor as constructed by the

Société de Laval (France) is shown in Fig. 110, and the parts are shown separately in Fig. 111. The half cylinders 8, 8, are

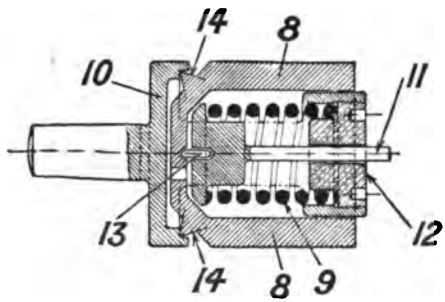


FIG. 110.—Section of Governor.

pivoted in the case 10 by the knife-edges 14, and have projecting lugs which press on the spindle 11 through the agency of pins 13.

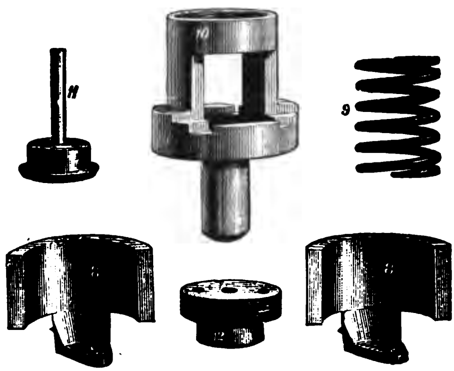


FIG. 111.—Parts of Governor.

Fig. 112 shows the half cylinders in their correct positions, but removed from the other parts. The spindle 11 acts by means of a lever on the steam admission valve. The centrifugal force is balanced by a spring, 9, which can be adjusted by means of the nut 12.

The connection of the governor with the steam admission or throttle valve, is shown in Fig. 113, where A is the spindle

strength is not required in these teeth, as the forces exerted on them are not excessive, owing to the high speed of rotation of the shaft A, and the small diameter of the pinion. It can be seen by a small calculation that, if the diameter of the pinion C be, say, one inch, and the speed of rotation 24,000 revolutions per minute, the force exerted on the teeth will be very small, even if the motor be of considerable power. D is the end bush of the turbine shaft, and F the middle bush, made in

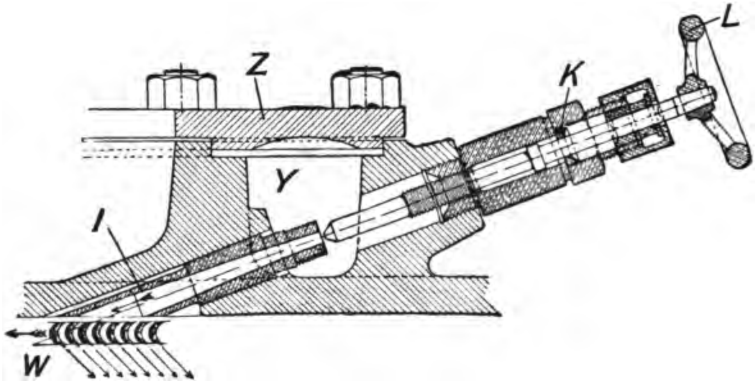


FIG. 109.—Nozzle and Vanes of a De Laval Turbine.

two parts. T is a tightening bush, also made in two parts. O, O, are the gear-wheel shaft bushes which support the power shaft N, which carries the gear wheel M, and the driving pulley R. S is a stop nut for the power shaft, and H a ball bush with adjusting spring for the turbine shaft. U is an adjusting nut, and V a friction gland. I is a steam nozzle, of which several are usually provided, distributed round the wheel. K is the stuffing-box for the spindle stop-valve, which can be actuated by the hand-wheel L. P is a lubricating ring, and Q is the governor which is mounted on the power shaft.

A section of a De Laval governor as constructed by the

Société de Laval (France) is shown in Fig. 110, and the parts are shown separately in Fig. 111. The half cylinders 8, 8, are

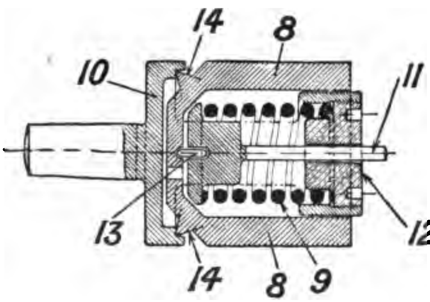


FIG. 110.—Section of Governor.

pivoted in the case 10 by the knife-edges 14, and have projecting lugs which press on the spindle 11 through the agency of pins 13.

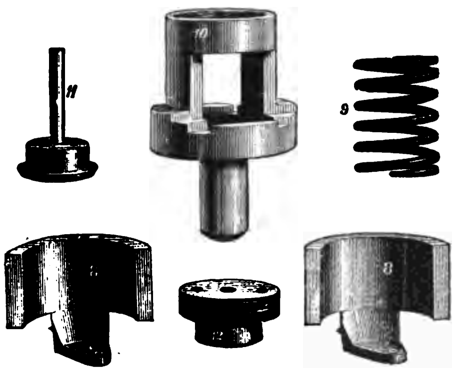


FIG. 111.—Parts of Governor.

Fig. 112 shows the half cylinders in their correct positions, but removed from the other parts. The spindle 11 acts by means of a lever on the steam admission valve. The centrifugal force is balanced by a spring, 9, which can be adjusted by means of the nut 12.

The connection of the governor with the steam admission or throttle valve, is shown in Fig. 113, where A is the spindle

which was marked 11 in Figs. 110 and 111. C is a lever pivoted near its centre, and arranged so that the spindle A can act on its lower end, while its upper end is connected to the lever G by means of a link which is adjustable by means of the nuts EF. The lever G operates the valve.

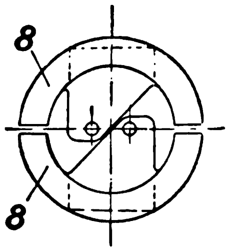


FIG. 112.—Half Cylinders of Governor in Position.

The characteristic feature of the De Laval steam turbine is the fact that the steam is completely, or nearly completely, expanded before it reaches the wheel. This expansion is accomplished by means of the divergent nozzle, which is best seen in Fig. 109. It will be seen from this figure that any nozzle may be closed by screwing down the spindle, and thereby preventing the entry of steam into the nozzle from the distribution conduit Y. This distribution conduit is cast in one of the parts of the casing in which works the turbine wheel, the conduit being closed by a ring, Z.

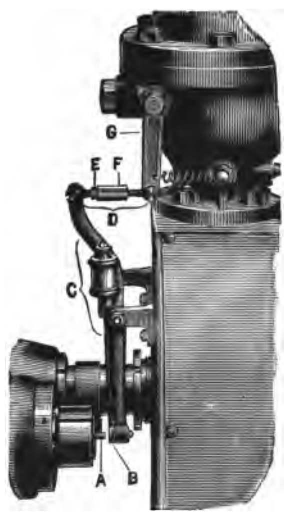


FIG. 113.—Connection of Governor with Steam Admission Valve.

The form of the nozzle I is most important. The sectional area of the smaller end has to be large enough to allow of the passage of the requisite amount of steam, while a sufficient section is required at the larger end for the complete expansion of the steam. The length of the nozzle must also exceed a certain amount, or the steam will take an eddying or irregular course through it. Too long a nozzle is objectionable, on account of friction.

This expansion of the steam before it enters the vanes is of great practical importance, for it allows of a considerable amount of clearance being permitted all round the turbine wheel in the case. In turbines constructed by the Société de Laval of France, a clearance of 2 to 5 millimetres is allowed all round the wheel. This permits of a very flexible shaft being used as a slight displacement of the wheel may take place without any injurious consequences. The pressure in the wheel-box or case is practically that of the exhaust exit from the case, and the turbine works like an impulse hydraulic turbine. The flexibility of the shaft is conducive to steadiness of motion at high speeds. To utilize a large proportion of the kinetic energy of the steam, as is done, the vane speed has to be enormous. Even the speed of the second motion or power shaft is usually very high. The great velocity and absence of pressure, however, allow of great lightness.

Table III. gives the total weights of steam turbines of various sizes as made by the Société de Laval, with the angular velocities of the second motion or power shafts. The first five sizes have each one power shaft; the others have two.

TABLE III.

WEIGHTS AND SPEEDS OF ROTATION OF DE LAVAL TURBINE MOTORS.

B.H.P. of turbine motor.	Total weight in kilograms.	Revolutions per min. of power shaft.
5	150	3000
10	225	2400
15	260	2400
20	420	2000
30	580	2000
50	1570	1500
75	1870	1500
100	2650	1250
150	3140	1040
200	4900	910
300	7650	775

The diameter of the turbine wheel is only 12 centimetres for a motor of 10 B.H.P., where the wheel revolves at the rate of 24,000 revolutions per minute, and 30 centimetres for a turbine of 100 B.H.P., with a wheel velocity of 15,000 revolutions per minute, while the turbine wheel, with an angular velocity of 7500 revolutions per minute, belonging to a 300 B.H.P. motor, has a diameter of 70 centimetres.

Plate III. shows 100 B.H.P. De Laval steam turbine dynamo made by Messrs. Greenwood and Batley, of Leeds. This machine has been delivered to, and is now at the works of, the Morris Aiming Tube and Ammunition Co., Ltd., Essex. It will be seen that there are two armatures. These are mounted on shafts which carry inside the gear-box helical-toothed wheels, which gear one on each side with a pinion mounted on the turbine shaft. The turbine wheel case is seen at the right of the figure, with the wheels for controlling the supply of steam to the several nozzles. The flange for connection to the steam supply is seen over the turbine case, and the exhaust outlet is shown at the bottom of the case. The weight of this machine complete is 6 tons, and the designed speed of rotation of the armature-spindles is 1050 revolutions per minute. A machine of the same power as this, but of earlier design, has been in constant use for several years at the Albion Works of Messrs. Greenwood and Batley.

Fig. 114 shows a De Laval turbine centrifugal pump as supplied by the same firm. T is the turbine-wheel casing, surrounding which can be seen the wheels W for controlling the steam jets. G is the gear-wheel casing, emerging from which can be seen the two pump shafts, which are driven like the armature shafts of the turbine dynamo just described.

Plate III.—100-B.H.P. De Laval Turbine Dynamo made by Messrs. Greenwood and Batley, Leeds.

Centrifugal pumps A and B are arranged one on each shaft, the two pumps working in parallel.

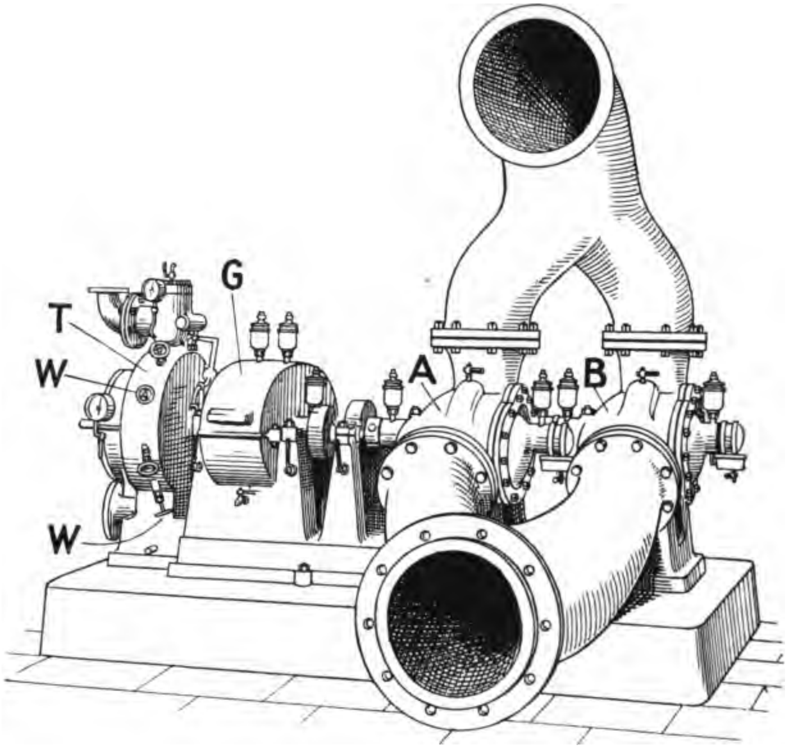


FIG. 114.—De Laval Turbine (Parallel) Centrifugal Pump.

Fig. 115 shows a turbine pump constructed by the Société de Laval, but in which the pumps are arranged in series as regards delivery of water. The pumps are arranged on parallel shafts as in the pump just mentioned, but in the series pump the delivery end of the one pump is connected to the suction end of the other, so that the pressure and not the amount of water is doubled.

The constant turning moment on the pump shaft causes

these pumps to work very quietly, and the high speed of rotation allows the water to be raised to considerable heights.

Fig. 116 shows a turbine blower as constructed by the French company.

Table IV. gives particulars of tests made by Messrs. Erik Andersson, Karl Wallin, and Axel Estelle, at the works of the Aktiebolaget de Laval's Angturbin in Sweden in 1895, on a 50 H.P. turbine dynamo. Steam was generated at 118 lbs. per square inch, and reduced by a throttle valve. The turbine had 6 induction nozzles.

TABLE IV.

TEST OF DE LAVAL TURBINE DYNAMO.

Date of trial.	E.H.P. $\frac{\text{volts} \times \text{amp.}}{736}$	Steam pressure; lbs. per square inch.	Vacuum; lbs. per square inch.	Number of nozzles used.	Lbs. of steam per E.H.P. per hour.
Feb. 15	49·92	114	13	6	24·5
Mar. 4	50·05	114	—	6	24·2
"	40·79	114	—	5	24·76
"	21·72	114	—	3	27·9
"	25·34	93·8	13·27	4	27·49
"	12·87	74	13·5	3	32·0

It should be noted that the electrical horse-power unit is obtained by dividing the product of volts and ampères by 736 instead of by 745, as is done in this country. The steam consumption was obtained on March 4, by inserting one of the steam nozzles into a pipe leading to a vessel containing a quantity of water where the steam was condensed. The amount of steam passing through this nozzle was thus ascertained, and it was assumed that the amount passing through each of the other '(acting) nozzles was the same, the design and cross-sections of the nozzles being identical. The amounts probably

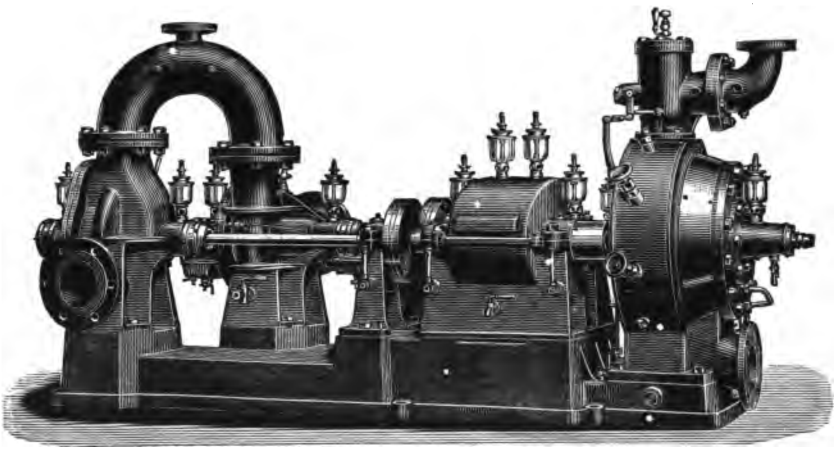


FIG. 115.—De Laval Turbine (Series) Centrifugal Pump.

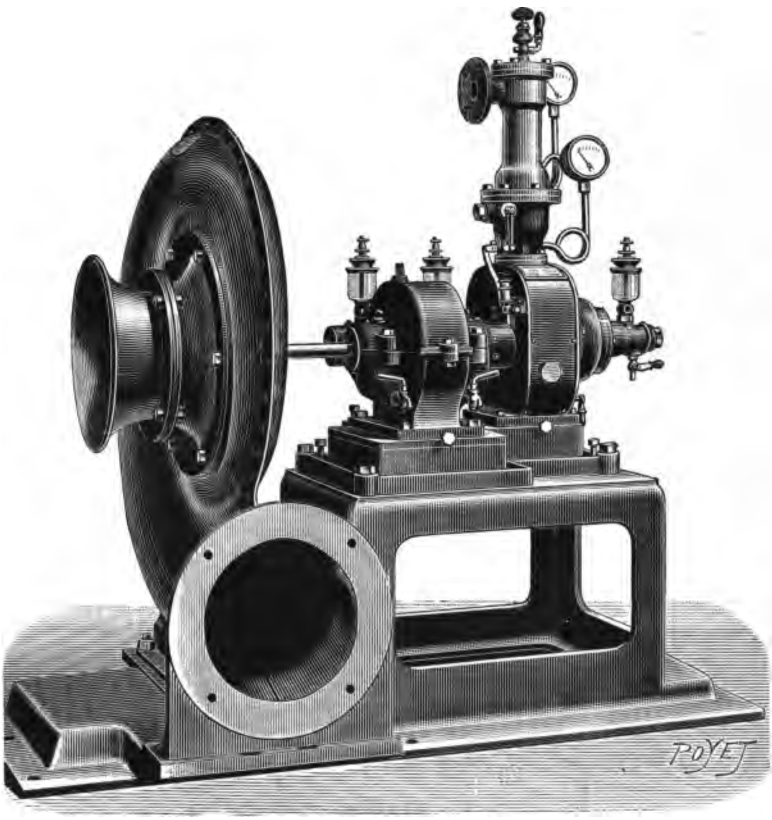


FIG. 116.—De Laval Turbine Blower.

were nearly the same, as was shown by a check test, but it cannot, of course, be assumed that this will be so in every case.

Table V. shows the results of tests on a De Laval turbine made by Professor Cederbolm, of Stockholm, in November, 1897. The power was measured by a brake.

TABLE V.

De Laval Turbine of 150 Brake Horse-Power.

No. of nozzles used.	Brake horse-power.	Steam pressure.		Vacuum.		Revolutions.	Consumption of steam per B.H.P. per hour.	
		Kilos per sq. centim.	Lbs. per square inch.	Millim. of mercury.	Inches of mercury.		Kilos.	Lbs.
7	165·3	8·00	113	670	26·4	1057	7·87	17·3
5	116·1	8·00	113	666	26·2	1057	8·01	17·6
3	65·0	7·90	112	685	27 0	1060	8·49	18·7

In 1896 tests were made of the steam consumption of one of the turbine dynamos supplied by the Société de Laval to the Edison Electric Illuminating Company of New York. The tests were made at the works of the Illuminating Company in New York.

The following is a summary of the trial, the report on which is signed by Messrs. Smith, Van Vleck, and De Kermel representing the Edison Electric Illuminating Company, and Mr. Paré, who represented the Société de Laval.

Duration of trial	6 hours.
Mean steam pressure	10 kilos. per square centimetre, or 143 lbs. per square inch.
Mean vacuum in condenser	65 centimetres, or 25½ inches.

Dynamo No 1, mean volts 127·25, mean ampères 708·56.
Dynamo No. 2, mean volts 128·26, mean ampères 727·47.

The total power generated was therefore about 183 kilowatts.

A surface condenser was used, and was tested to prove that no leakage took place. The amount of water condensed in the six hours was 12,493·35 kilograms, or 2082·22 kilograms per hour.

The steam consumption per kilowatt hour was therefore $\frac{2082 \cdot 22}{183}$, or 11·38 kilograms (*i.e.* 25·1 lbs.).

Seven hundred and thirty-six watts were taken as an electrical horse-power, and the efficiency of the dynamo was assumed to be 90 per cent. The brake horse-power of the turbine under these assumptions was therefore about 276·7, and the consumption of steam per B.H.P. per hour $\frac{2082 \cdot 22}{276 \cdot 7}$, or 7·52 kilograms (*i.e.* 16·6 lbs.).

CHAPTER IX.

THE RATEAU STEAM TURBINE.

It has already been pointed out that in the Parsons turbine the steam is expanded gradually in passing alternately through fixed and moving rings of blades, while in the De Laval type of turbine, the steam is expanded in a divergent nozzle before it reaches the vanes of a single rotating wheel. It has also been pointed out that the De Laval type of motor has an advantage over the Parsons type in so far that the amount of clearance round the wheel does not need to be small. In a Parsons turbine, the amount of clearance spaces between the fixed and moving rings of blades have to be most minute to prevent excessive leakage of steam, especially at the high-pressure end of the turbine. It is true that the leakage of steam round the rings of blades, instead of through between the blades, does not represent the same loss of power as the leakage of steam past the piston of a reciprocating engine, for the steam that leaks past one ring of blades reserves its energy for the next ring, or gives up its saved heat to the rest of the steam. It will, however, be obvious on a little consideration that this leakage of steam must entail a lengthening of the turbine cylinder, and an increase in the number of rings of blades, in order to expand the steam to the desired extent. This involves increase in bulk, weight, cost, and radiation.

To minimize the leakage of steam, great accuracy and good workmanship are required; and, although these requisites can be commanded by Messrs. C. A. Parsons and Co., they might not be obtained in less well-equipped or less well-managed works.

The disadvantage of the De Laval type of steam turbine is the excessive velocity which the blades must have, necessitating the use of gearing to obtain speeds of rotation which can be utilized for industrial purposes. In the more powerful De Laval motors, the larger turbine wheels employed allow somewhat smaller angular velocities to be obtained without reducing the velocity of the vanes; but in all cases the number of revolutions per minute of the turbine wheel is very high. Apart from the objectionable feature of gearing, the velocity of the vanes is limited by the strength of the materials of construction. As has already been pointed out in Chapter V., the stress due to centrifugal force in a rotating ring becomes enormous at high velocities. The vane-speed in a De Laval turbine is thus limited by the strength of material obtainable. Increasing the diameter of the wheel makes approximately no difference if the vane-speed remains constant. By arranging the vanes on the periphery of a disc which increases in thickness from the circumference to the centre, a somewhat higher speed may be obtained, the inner parts of the disc supporting the outer, but still a limit is reached to the safe speed before the best velocity is obtained for utilizing the energy of high-pressure steam.

The Rateau steam turbine, as now constructed by Messrs. Sautter, Harlé and Co., of Paris, and by the Maschinenfabrik Oerlikon of Switzerland, has been devised with the object of obtaining the advantages of the De Laval motor while adopting

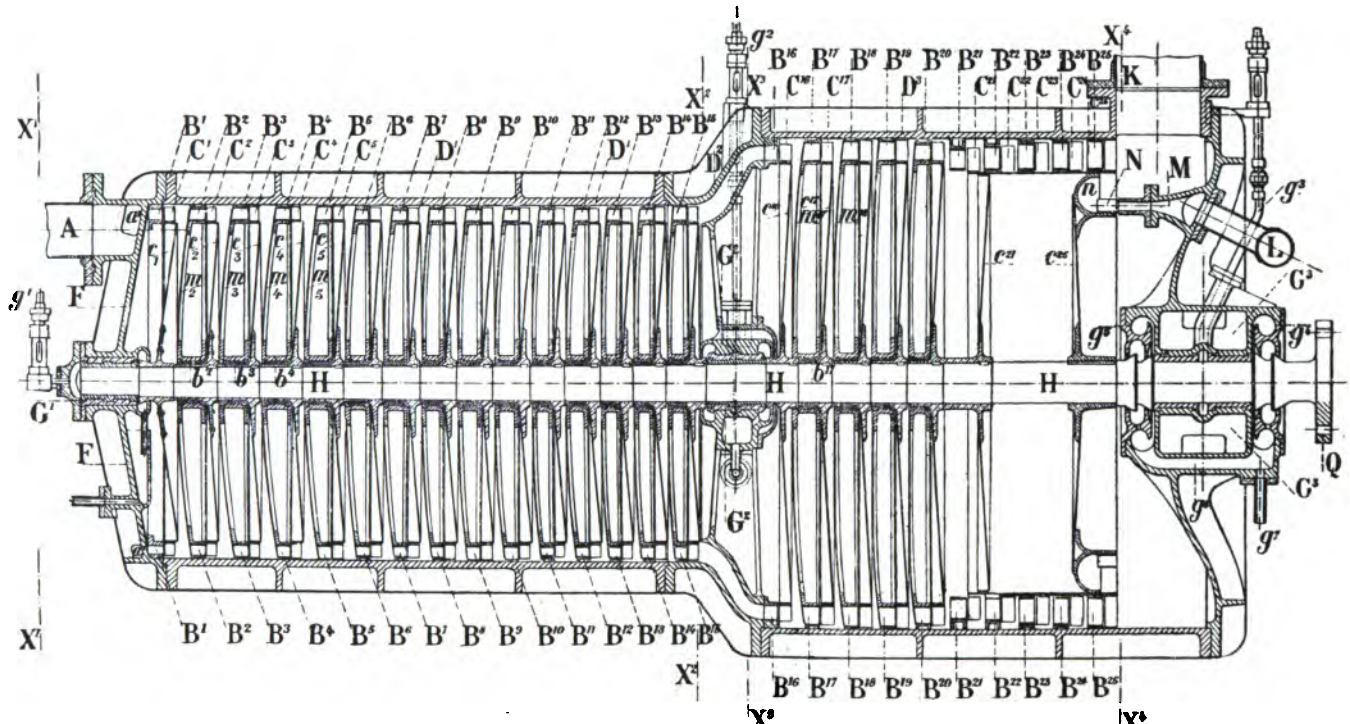
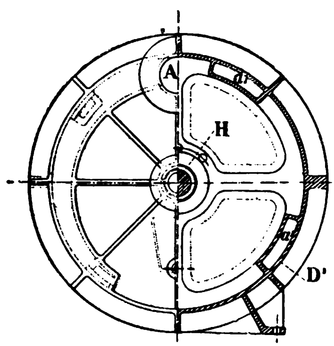


FIG. 118.—Rateau Steam Turbine: Longitudinal section.

the plan of expansion in steps and action on a series of wheels in order to obtain a more moderate speed of rotation.

This type of steam turbine is somewhat like the Parsons parallel flow-motor, but differs from the latter in this respect, that each rotating ring of blades revolves, as it were, in a compartment by itself. If we can imagine a number of De Laval turbines placed side by side with the wheels in parallel planes, and if we imagine a large number of nozzles extending from the exhaust side of one wheel, through the casings to the steam side of the next wheel, we have in principle a Rateau turbine as now constructed at Oerlikon and by Messrs. Sautter, Harlé and Co., of Paris.

Fig. 118 is a longitudinal section through a marine steam turbine of this type, and Figs. 119 and 120 are transverse



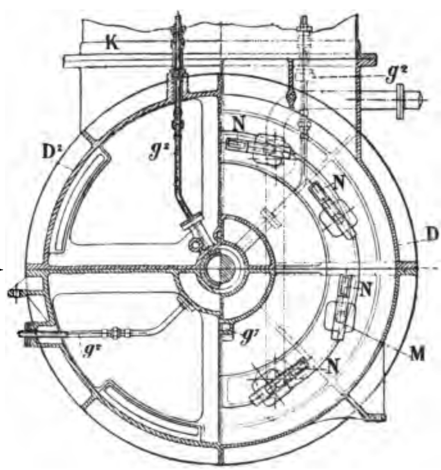


FIG. 119. FIG. 120.

Transverse Sections of Rateau Steam Turbine.

sections of the same. The plane of section of the left-hand half of Fig. 119 is represented by the line X^1X^1 on Fig. 118, and the plane of section of the right-hand half by the line X^2X^2.

The planes of section of the left and right-hand parts of Fig. 120 are represented on Fig. 118 by the lines X^3X^3 and X^4X^4 respectively.

The cast-iron or cast-steel cylinder D^1, D^2, D^3 is made in several parts and is strengthened by circumferential ribs. The high-pressure end of the cylinder is closed by the dished plate F to a flange on which is attached the steam-pipe A. Steam passages, a^1, a^1, are provided to allow of the steam reaching the first distributor or guide ring B^1. This distributor consists of a series of blades which occupy a portion of the inner circumference of the casing. These blades guide the steam in the proper direction on to the blades C^1 of the first rotating disc c^1. This disc is of thin steel slightly dished and is attached to an annular flange formed on a hub mounted on the turbine shaft H. The disc is formed with a circumferential flange to which the blades are attached. Fig. 121 shows a method of riveting the rotating blades C to the flanged periphery P of the disc, two consecutive blades being shown. The pieces U are cast on to the blades at their flanged ends to stiffen them. It will be seen from the figures that the arrangement is very light. The steam on passing the rotating blades C^1 enters a chamber enclosed between the disc c^1 and a diaphragm m^2. This diaphragm extends from a hub, b^2, which surrounds the shaft without touching it to the distributing blades B^2. These blades are fixed to a casting which is attached steam-tight to the inside of the cylinder. The construction is such that the steam can enter the chamber only by way of the rotating blades C^1, and can leave the chamber only by way of the distributing blades B^2. These distributing blades direct the steam on to the second set of rotating blades C^2, after passing

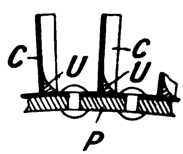


Fig. 121.

through which the steam enters another chamber enclosed between the disc c^2 and diaphragm m^3, which is attached to the distributing blades B^3 and to the hub b^3, the diaphragm, distributing blades, and hub being similar to the preceding, except that the area allowed for the passage of steam is greater. The construction is continued in a similar manner to the end of the cylinder. The diameter of the cylinder is increased at D^2 to afford greater area to the steam. Any steam that may leak out between any set of distributing blades and the succeeding rotating blades is in a closed chamber between the diaphragm of this set of distributing blades on the left hand and the diaphragm of the next set of distributing blades on the right hand. Each revolving disc, therefore, rotates in a closed chamber just like a De Laval turbine, the pressure on both sides of the rotating disc being approximately the same. There is, of course, a slight leakage of steam between the hubs b^2, b^3, etc., and the shaft; but a clearance of a few millimetres here does not allow a large area for the escape of steam, and any distortion of the machine is not so likely to cause rubbing at this point as at the circumference of the rotating discs.

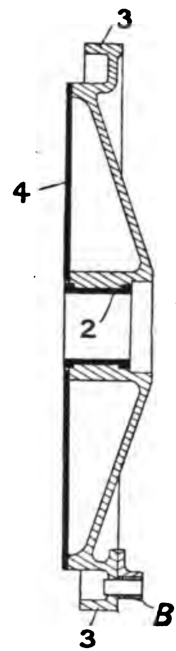


FIG. 122. — Diaphragm and Distributing Vane of Rateau Turbine.

Fig. 122 shows one of the diaphragms attached to the distributing blades of a slightly different design to that shown in Fig. 118. The bush 2 inserted in the hub is just clear of the shaft. The part 3 fits into a groove in the surrounding cylinder. One of the distributing vanes is

shown at B, these vanes being usually fitted only on a small part of the circumference at the high-pressure end of the turbine and increasing in number towards the low-pressure end; 4 is a plate riveted on the front of the diaphragm in order to present a smooth surface to the steam and so reduce friction.

The last five rings of rotating blades C^{21} to C^{25} are not mounted like the others, but are attached to the exterior of a drum which is connected to the shaft H by the discs c^{21} and c^{25}. The distributing blades B^{21} to B^{25} are connected only to the enclosing cylinder. K is the exhaust passage, and it will therefore be seen that the back of the plate c^{25} is exposed to the pressure of the exhaust, while the front of the plate c^{21} is exposed to the pressure of the steam which acts on the blades C^{21}. An axial thrust is thus exerted on the rotating parts of the turbine, and this axial thrust is used to wholly or partially balance the thrust of the screw propeller. In a turbine used for driving a dynamo or otherwise where no axial thrust is required, this arrangement of blades which is the same as that in a Parsons parallel-flow turbine may be dispensed with. The arrangement has the disadvantage, mentioned at the beginning of this chapter, that leakage of steam will take place between the inner periphery of the distributing rings of blades and the exterior of the drum carrying the rotating blades.

For rotating the turbine in a reverse direction (when this is required) a number of vanes, N, are provided, the curvature of which is opposite to that of the other moving vanes. Steam is guided on to these vanes by nozzles, M, leading from a supply conduit L. The rotating vanes, N, are carried by the disc c^{25}, and the steam exhausting from these vanes is guided by the annular trough *n* to the exhaust passage K.

The shaft K is supported in three bearings G^1, G^2, G^3. These bearings are supplied with oil under pressure conveyed to the respective bearings by the pipes g^1, g^2, g^3. The pressure of oil in the bearing G^3 is used to prevent air leaking into the exhaust end of the turbine when the latter is connected to a condenser.

M. Rateau has also experimented, in conjunction with Messrs. Sautter, Harlé and Co., with other types of steam turbines having one, two, or more discs. Some account of these experiments is given in M. Rateau's interesting paper presented to the International Congress on Applied Mechanics held at Paris in the summer of 1900. In a one-disc turbine described in the paper, the disc was formed from a single piece of special forged steel in the periphery of which the vanes were milled. These vanes were of the double type like those of a Pelton water-wheel. The disc increased in thickness from the periphery to the centre, this design being adopted for the sake of strength to resist centrifugal force. The stress produced in a ring due to the centrifugal force caused by its own weight, has already been discussed in Chapter V. M. Rateau has calculated that, by substituting a disc of uniform thickness for a ring, the allowable speed of rotation is only increased by 7 per cent. In this case the dangerous part is at the centre of the disc. By increasing the thickness progressively towards the centre, M. Rateau has found that considerably higher speeds can be attained. In fact, with special hard steel he has obtained, without rupture, peripheral velocities of 400 metres per second.

The steam was projected on to the disc by several nozzles, and the jets of steam were divided in two by the central ridges of the buckets, as in the Pelton water-wheel. The nozzles

were arranged to project the steam on to the lower part of the disc so that the impact of the steam helped to balance the weight of the disc. One or more of the nozzles could be put out of action to decrease the power of the motor. The best results were obtained by supporting the disc shaft in one bearing only. The disc had thus a slight play owing to the flexibility of the shaft, and was able to choose its own centre of rotation. Gearing was employed to reduce the speed of the disc, the gear-wheels being of double helicoidal form and enclosed in a dust-proof box. The best form of packing tried at the place where the shaft passed through the side of the casing containing the disc consisted of a ring split into three pieces along three diametrical planes. This split disc was pressed against the side of the casing by means of springs. When the shaft did not vibrate, the ring worked as if solid, whilst, when the shaft did vibrate, the three pieces moved apart to give it freedom. This packing was found to be tight as long as the vibrations of the shaft were not considerable.

CHAPTER X.

FURTHER REMARKS ON THE PARSONS TURBINE.

THE efficiency of a condensing steam turbine depends largely on the pressure at the exhaust end, or, in other words, the number of inches of vacuum at this end. Tables VI. and VII. show the effect on a Parsons turbine of altering the vacuum in the condenser. It will be seen that every additional inch of vacuum reduces the steam consumption about 4 per cent.

TABLE VI.

CONSUMPTION OF 500-KILOWATT PARSONS TURBO-ALTERNATOR RUNNING AT 2500 REVOLUTIONS WITH 140 LBS. STEAM PRESSURE AT THE STOP-VALVE AND NO SUPERHEAT. (Based on results of tests.)

Vacuum constant from full load to no load.	Consumption of steam per kilowatt-hour.			Consumption per hour.
inches of mercury.	full load.	half load.	quarter load.	no load,
29	—	—	—	1500
28	22·2	25·6	32·4	1700
27	23·1	26·9	34·5	1900
26	24·0	28·2	36·6	2100
25	25·1	29·7	39·0	2300
24	26·2	31·2	41·2	2500
23	27·5	32·9	44·8	2700
22	28·9	34·7	46·3	2900

Barometer = 30 ins.

TABLE VII.

CONSUMPTION OF 1000-KILOWATT TURBO-ALTERNATOR CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO. FOR ELBERFELD CORPORATION. NO SUPERHEAT.

Pressure stop-valve.	Vacuum. Barometer = 30 ins.	Kilowatts.	Steam per kilowatt-hour.
lbs. per sq. in.	inches of mercury.		lbs.
157·5	26·97	1010	23·08
153·0	24.45	1041	25·25
125·0	27·10	1022	20·47

But even with a good vacuum in the condenser, the efficiency may be spoilt by the throttling of the exhaust by narrow pipes or passages between the turbine and the condenser. To prevent any possibility of this, Mr. Parsons has invented

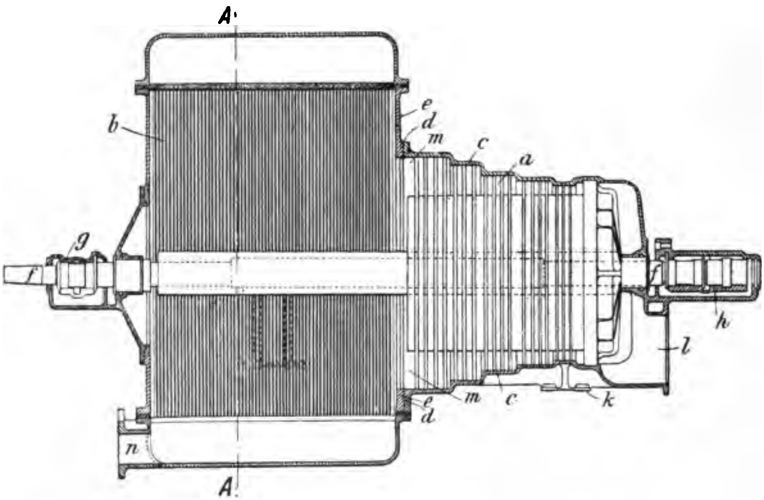


FIG. 123.—Vertical section.
Parsons Combined Turbine and Condenser.

and patented a combined turbine and condenser, which is illustrated in Figs. 123, 124, and 125, the turbine being of the parallel-flow type. Fig. 123 shows the combination in

vertical section, and Fig. 124 in plan; while Fig. 125 is a partial vertical section on the line AA of Fig. 123. The casing *c* of the turbine is bolted at *d* to the end *e* of the condenser.

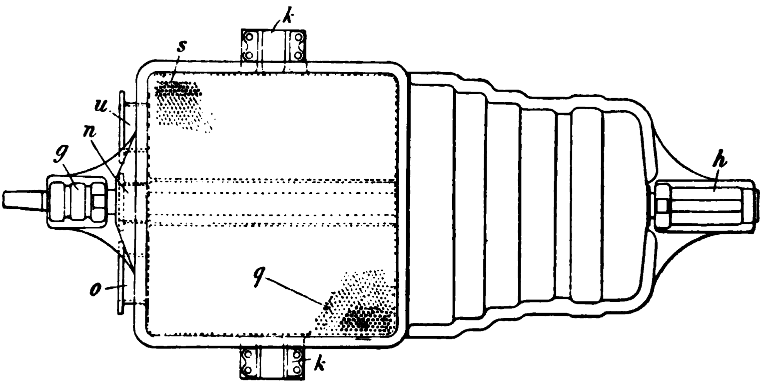


FIG. 124.—Plan.
Parsons Combined Turbine and Condenser.

The turbine spindle *f* passes through the turbine and condenser, and is supported in bearings at *g* and *h*. The turbine and condenser casings are supported on feet, *k*, *k*. The steam enters the turbine casing at *l*, and, after going through the fixed and moving rings of blades *a*, passes directly out of the large end of the turbine casing into the condenser. The turbine and condenser casings act as if made in one piece, and are, in fact, only made in separate castings for constructional reasons. The outlet from the condenser to the air-pump is shown at *n*. The circulating water enters the condenser at *o*, and passes into compartment *p*. It then passes up through the tubes *q* to the

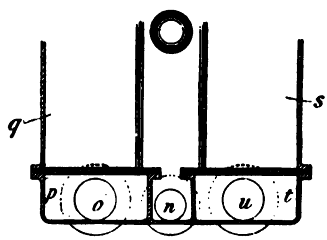


FIG. 125.—Partial vertical section on line AA of Fig. 123.
Parsons Combined Turbine and Condenser.

top chamber *r*, whence it descends through the tubes *s* to the compartment *t*, and leaves the condenser at *u*.

When it is desired to cause the shaft of a turbine to revolve in a reversed direction, this is usually accomplished by placing a reversing turbine on the same shaft as the main turbine. The main and reversing turbines are usually in separate casings, and steam is admitted to one or the other according to the direction of rotation desired. Both have their exhaust ends permanently connected to the condenser, so that the one not working rotates in the condenser vacuum; and, as there are no rubbing parts within the casing of a turbine, the drag of the inoperative turbine is almost inappreciable.

The main and reversing turbines may, however, with advantage be placed within the same casing. Fig. 126 shows

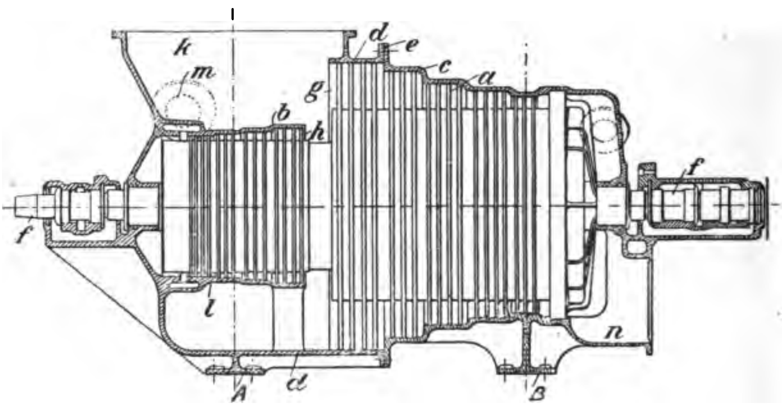


Fig. 126.—Parsons Arrangement of Main and Reversing Turbines in One Casing.

an example of this. The main turbine *a* is enclosed chiefly in the casing *c*, which is bolted at *e* to the casing *d*. The casings *c* and *d* are cast with feet, A and B, and the casing *d* also carries an internal cylinder, *l*, which encloses the reversing turbine *b*. Both turbines are of the parallel-flow type, and both have their

moving rings of blades attached to the spindle *f*. The low-pressure ends *g* and *h* of the two turbines open into the passage *k*, leading to or forming a part of the condenser. The steam supply for the main turbine enters by the passage *n*, while that for the reversing turbine is admitted through the casing *d* at *m*. The turbines shown in Fig. 126 are intended for marine purposes, and the reversing turbine is therefore smaller than the main turbine, as the astern speed of a vessel is not usually required to be so great as the ahead speed.

Fig. 127 shows another example of main and reversing

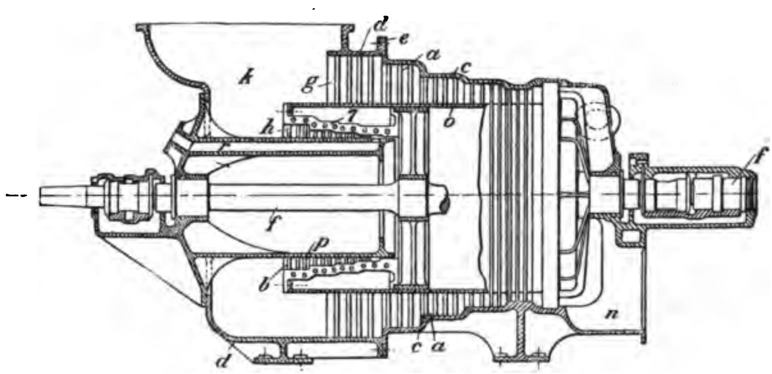


Fig. 127.—Parsons Arrangement of Telescoping Reversing Turbine within Main Turbine.

turbines in one casing. The reversing turbine *b* is here telescoped within the main turbine to save longitudinal space. The stationary rings of vanes of the main turbine are fixed, as is usual, to the casing *c*, the moving rings being attached to the drum *o*, which is fixed to the shaft *f*. The reversing turbine has its fixed rings of blades attached to the exterior of the cylinder *p*, which is fixed to the casing *d*, while its moving rings are carried by the casing 7, which is rigid with the drum *o*. The steam enters the main turbine at *n*, while

it gains access to the reversing turbine by the pipe *r*. The exhaust ends, *g* and *h*, of both turbines open directly into the condenser passage *k*.

A Parsons turbine can be reversed by interchanging its steam and exhaust connections so that the steam passes through the turbine in the reverse direction, but the efficiency is not as great. If the blades are designed for maximum efficiency in one direction, the efficiency when rotating in the other direction is much reduced. The usual construction of blades has been already shown in Figs. 3, 4, 5, 8, 10, and 72A. It will be seen that both in the fixed and in the moving blades the space between two adjacent blades converges from the side at which the steam enters to the side at which it leaves. The concave face of the blade at the side at which the steam enters is almost at right angles to the direction of motion of the moving blades. When the flow of steam is reversed, the blades are much less efficient. Mr. Parsons has patented the form of blade illustrated in Fig. 128 for use in turbines intended to run in

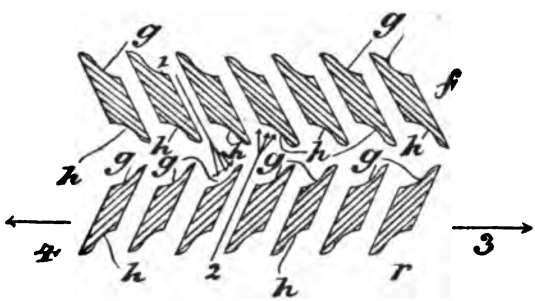


FIG. 128.—Form of Blades adapted for Rotating in Either Direction.

both directions. Here the blades are straight for the greater part, but each blade is hollowed out at both ends, at *g* and *h*, so that, whichever way the steam flows, it impinges on a concavity. The fixed blades are lettered *f*, and the revolving

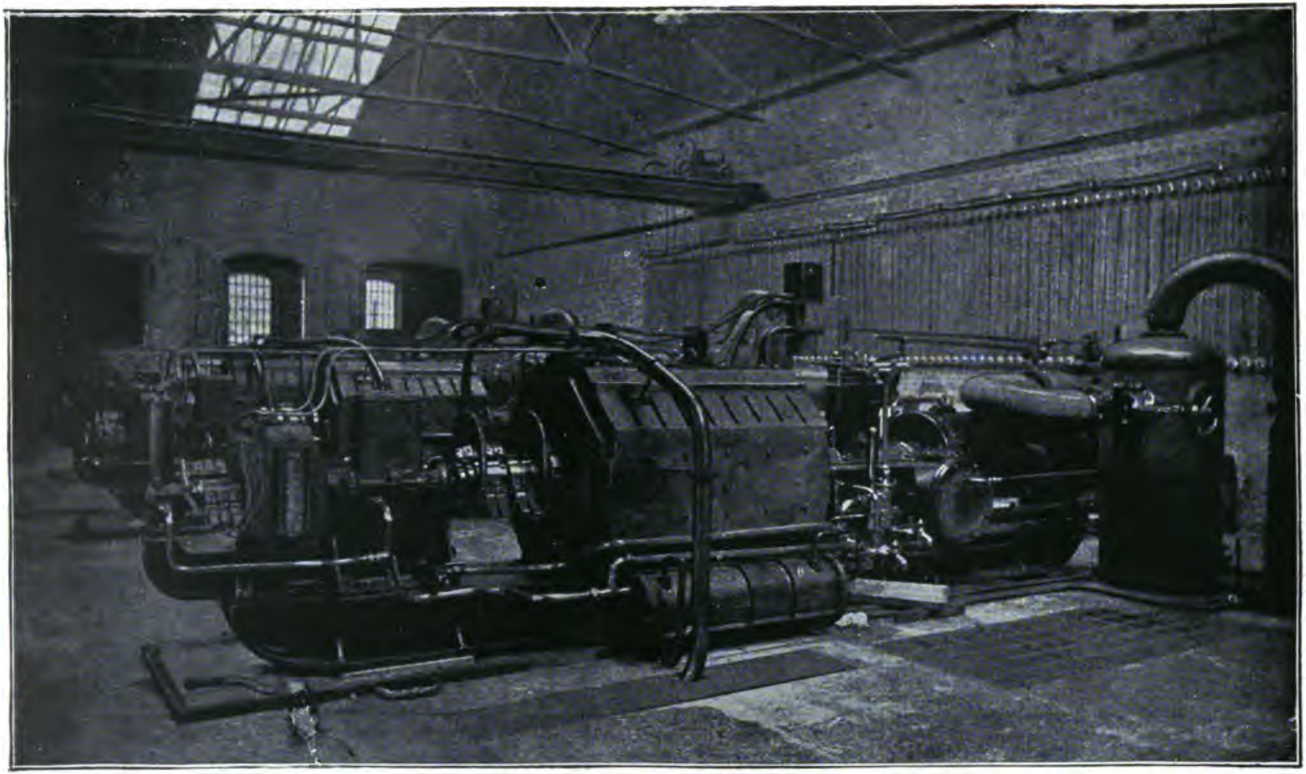


Plate IV.—Metropolitan Electric Supply Company's Sardinia Street Station with Parsons Steam Turbines coupled to Alternators.

blades *r*. The latter move in the direction of the arrow 3 when the steam passes as indicated by the arrow 1, and in the direction of the arrow 4 when the steam flows as indicated by the arrow 2. Messrs. C. A. Parsons and Co. have a very ingenious machine for constructing the rings of blades used in their turbines. Shrouds of suitable metal, preferably brass, are formed into a circle, or segment of a circle. On one edge of the strip, teeth of special shape are cut by means of a circular cutter. The form of the teeth is such that, when the blades are laid in the grooves and the teeth turned over them, the teeth and blades fit each other closely, and form a secure fastening. This will be clear by referring back to Figs. 3, 4, and 5. In Fig. 3 some of the teeth of the shrouds are shown before they are bent down over the blades. The bending down of the teeth is performed by a punch which acts about three or four teeth behind the cutting-tool, so as to give the attendant time to insert the blades. The rings of blades are usually constructed with a heavy shroud at one end and a light shroud at the other, the heavy shroud being inserted and caulked into a groove in the turbine casing or revolving drum.

The governing of a Parsons turbine is usually effected by varying the duration of puffs or blasts of steam admitted to the turbine. Fig. 129 shows an electrical governor arranged for this purpose. The solenoid U is energized by electric current (from the electric generator driven by the turbine), so that increase or decrease of speed of the turbine causes the lever U^2 to overcome the resistance of the spring U^1, or to be overcome by it. This lever, by means of the projection U^3, moves a cam sleeve, V, on the second-motion shaft Q^1. The sleeve, although free to slide along the shaft, rotates

with it, and the cam surface cut on the sleeve acts on a roller so as to depress the steam-valve spindle R against the spring R^1. The cam surface is so arranged that in one position of the sleeve V, the steam-valve is held open during the whole revolution of the shaft Q^1—that is, steam is admitted continuously by the steam valve to the turbine. When, however,

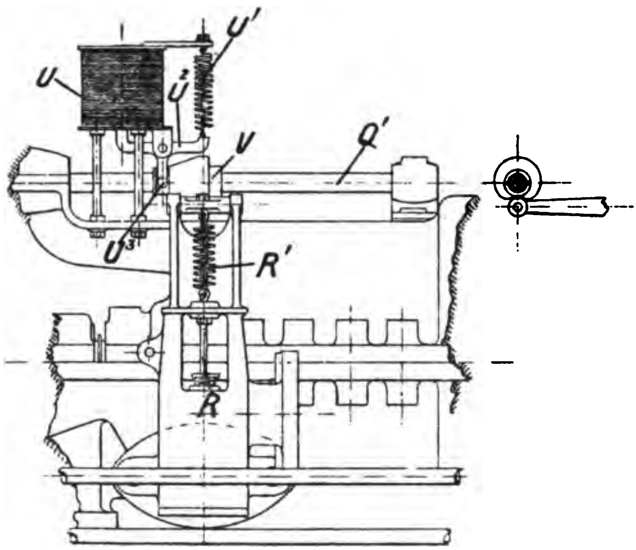


Fig. 129.—Electrical Governor for Parsons Turbine.

the solenoid gets more strongly energized, it pulls the sleeve V to such a position that steam is admitted to the turbine only for a portion of a revolution of the shaft Q^1; and, the greater the energizing current, the further the sleeve moves along, so that steam is admitted to the turbine for a smaller and smaller fraction of a revolution of the shaft Q^1. The shaft Q^1 is driven by the frictional contact of a wheel or disc carried by it with the end of the turbine spindle, the speed of revolution of the shaft Q^1 being much less than that of the turbine spindle.

Figs. 130 and 131 show another arrangement of electric governor. Steam is admitted to the turbine by the double-beat valve A from the steam space A^1. Steam which leaks past the neck-bush C acts on the piston B^1 so as to force it upwards against the action of the spring B^2. This occurs when the valve D is closing the passages B^3 and B^4, but, when these passages are opened, the steam escapes from the lower end of the cylinder faster than it can enter by the leak; and so the piston B^1 descends, and, by means of the rod B, closes the main valve A. For intermediate positions of the valve D, the main valve assumes positions of partial opening. The eccentric G^1 is driven from the turbine spindle H by means of a worm and worm-wheel, and gives a rocking motion to the lever G. This is pivoted at G^2, and, consequently, its end E^1 has an up-and-down motion. This end, E^1, is connected to a lever, E, one end, E^2, of which is attached to the valve D. The lever E can turn about the point E^3, and the valve D will, therefore, be reciprocated up and down by the action of the eccentric G^1. This will allow regular puffs or blasts of steam to pass through the valve A.

The other end of the lever E is pivoted to the core F^1 of the solenoid F, which tends to draw it down against the action of a spring at E^3; so that an increase or diminution in the strength of the current energizing the solenoid will cause the lever E to turn about the point E^1 and actuate the valve D. The effect of the combined action of eccentric and solenoid is to prolong or shorten the duration of the puffs, and the turbine is thus governed.

Centrifugal governors may be employed to control the admission of steam equally as well as electrical governors. For example, in Fig. 129 the sleeve V might have been actuated

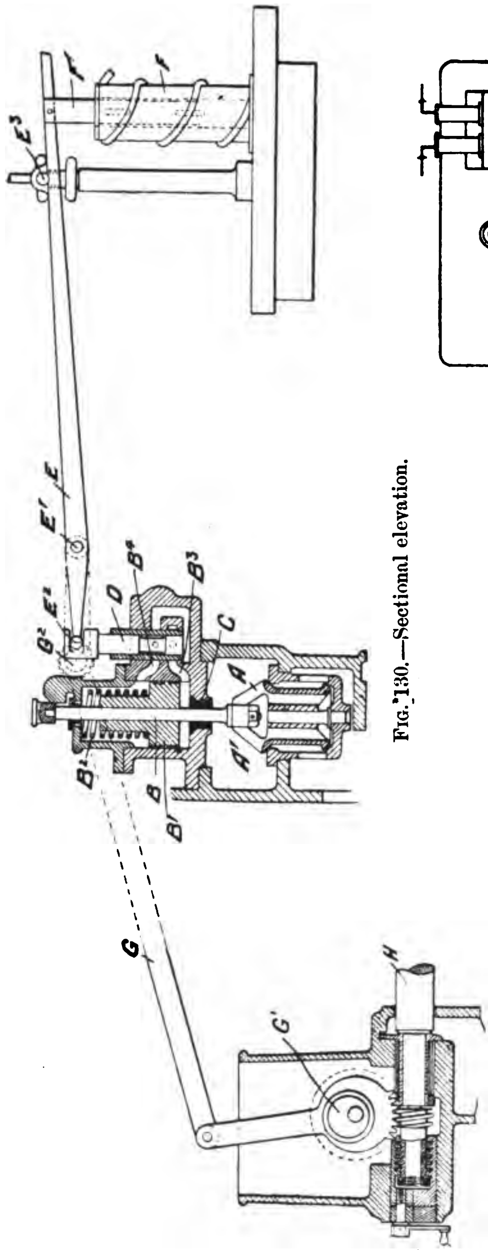


FIG. 130.—Sectional elevation.

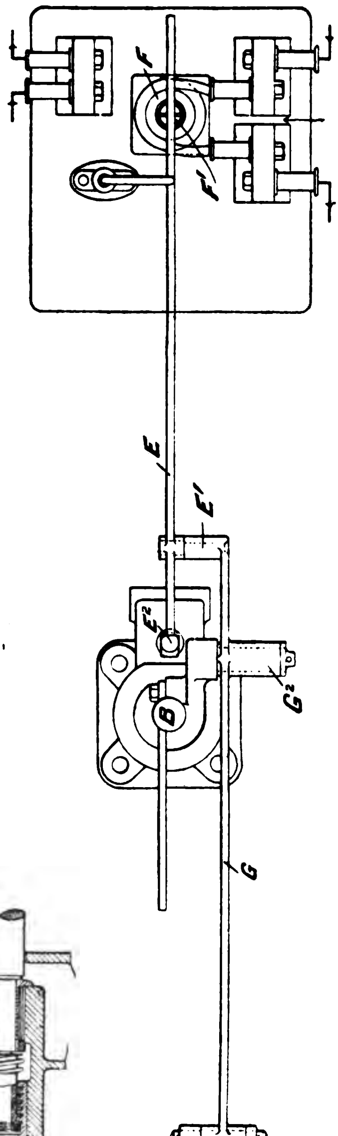


FIG. 131.—Plan.
Electrical Governor for Parsons Turbine.

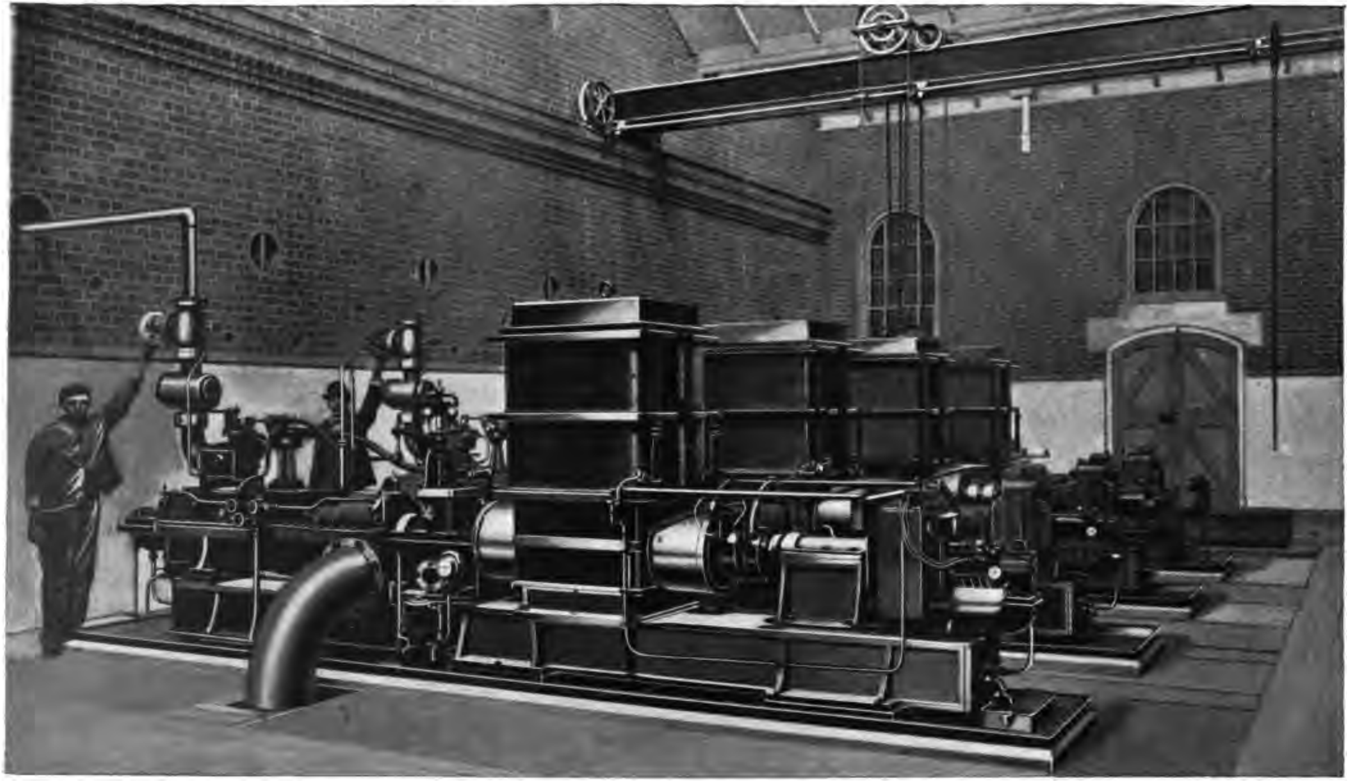


Plate V.—Victorian Railways Lighting Station equipped with Four 150-Kilowatt Parsons Turbine Alternators with Exciters.

by a centrifugal governor mounted on and rotating with the second-motion shaft Q^1. The electrical governor has the advantage in cases where constant voltage is required, as it can control the voltage independently of the speed.

Plate IV. shows an installation of steam turbine alternators supplied by Messrs. C. A. Parsons and Co. to the Metropolitan Electric Supply Co. An injunction was obtained against this company to cease running at their Manchester Square Station unless the vibration was prevented. The company satisfied the plaintiffs by removing the reciprocating engines then installed, and replacing them by steam turbines.

In Plate V. is seen the interior of the Victorian Railways Lighting Station. Four turbine alternators, each of 150 kilowatts capacity, and provided with exciters, are there installed and run in parallel. Ferranti rectifiers are used, and the current employed for both arc and incandescent lighting.

Plate VI. shows a steam turbine driving a centrifugal pump. This was supplied by Messrs. C. A. Parsons and Co. to Messrs. Storey Bros., Lancaster, and is said to be capable of delivering 53,000 gallons per hour against a head of 165 feet.

Parsons turbines are also used for directly driving fans and air-propellers. Plate VII. shows a turbine-driven fan installed at the Clara pit in Durham. It is 5 feet in diameter, and said to be able to deliver 120 cubic feet of air per minute at a pressure equal to 2 inches of water.

CHAPTER XI.

SOME RECENT TESTS OF PARSONS TURBINES.

THIS chapter will be devoted to giving the results of some recent tests of Parsons turbines.

TABLE VIII.

TEST OF 24-KILOWATT TURBO-DYNAMO FOR MESSRS. SPILLERS AND BAKERS, NEWCASTLE-ON-TYNE, CONSTRUCTED BY MESSRS. C. A. PARSONS AND CO.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. Bar.=30″.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
80	0	28·8	4990	24·7	712	28·8
77	0	29·0	4630	11·8	400	33·9
74	0	29·1	4570	5·15	235	45·6
78	0	26·0	4900	23·8	798	33·5
79	0	0	4780	19·7	1350	68·5

This shows that good efficiencies can be obtained even with steam turbines of comparatively small size. It also shows the effect of better vacuum and higher load on the steam consumption. The former was also shown by Tables VI. and VII., pp. 115 and 116.

TABLE IX.

50-KILOWATT STEAM ALTERNATOR SUPPLIED BY MESSRS. C. A. PARSONS AND CO. TO THE BLACKPOOL CORPORATION.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. Bar.=30″.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
126	0	28·0	5044	52·7	1480	28·0
132	0	28·5	4880	0	320	—

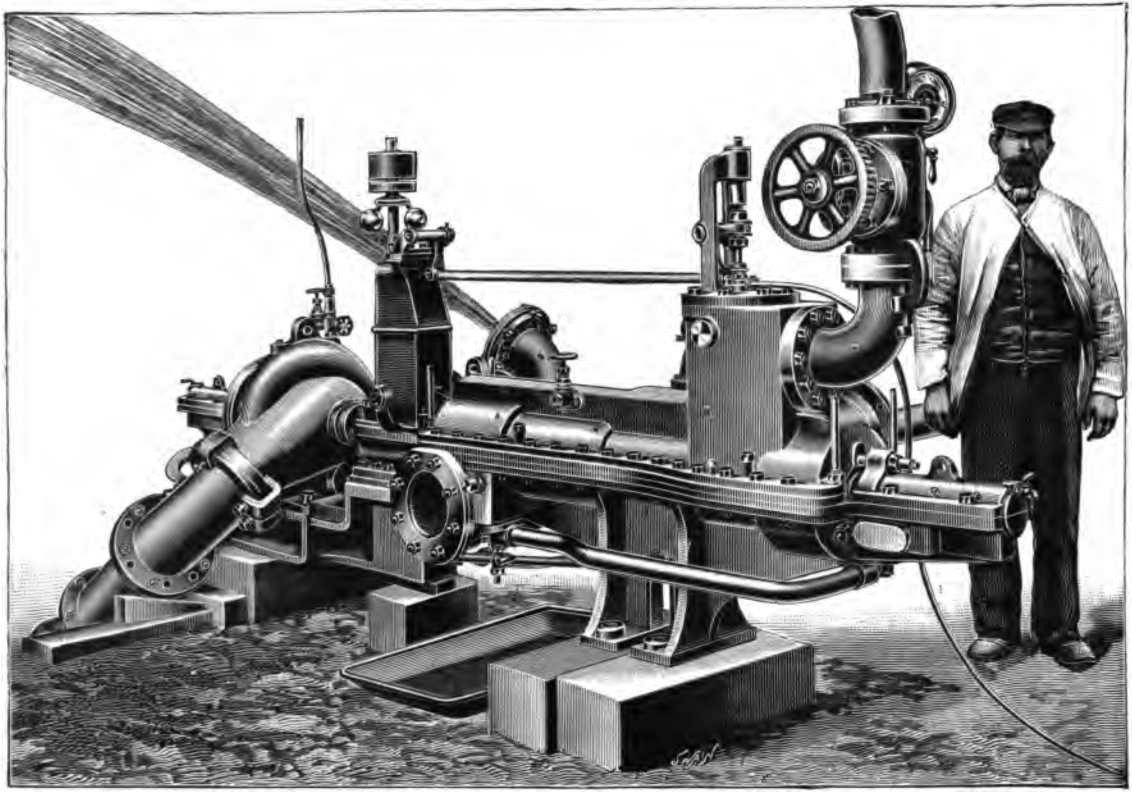


PLATE VI.—PARSONS STEAM TURBINE COUPLED TO CENTRIFUGAL PUMP.

With a larger power and higher steam-pressure the efficiency here is slightly greater.

TABLE X.

TWO 100-KILOWATT CONTINUOUS-CURRENT TURBO-DYNAMOS FOR WEST BROMWICH ELECTRIC LIGHTING STATION, MADE BY MESSRS. C. A. PARSONS AND CO.

Pressure of steam above atmosphere at stop-valve.	Superheat at stop-valve.	Vacuum in the turbine cylinder. Bar.=30″.	Revolutions per minute.	Load.	Steam used.	
lbs. per sq. in.	degrees F.	ins. of mercury.		kilowatts.	lbs. per hr.	lbs. per kw.-hr.
129	54	27·8	3500	123	3144	25·5
134	64	27·7	3520	122	2913	23·8

With a greater power and moderate superheat the efficiency is again improved.

In January, 1901, a series of trials were made by Professor Ewing of a 500-kilowatt steam turbo-alternator at the works of the Cambridge Electric Supply Co.

The machine was constructed by Messrs. C. A. Parsons and Co., and erected at the Cambridge Electric Co.'s station in January, 1900, and ran at times daily, and at times intermittently, according to requirements, up to the time it was tested.

The turbine is of the parallel-flow type, with its shaft as usual directly coupled to the armature of the alternator, which is of the four-pole type, and designed to give 250 ampères at 2000 volts, running at 2700 revolutions a minute. The turbine is governed electrically, and is furnished with a surface condenser, and drives its own air-pump and circulating pump by means of a shaft carrying a screw-wheel driven by a worm on the main turbine shaft.

Table XI. shows the collected results of the trials, and Figs.

132 and 133 show the steam consumption graphically. The straightness of the line in Fig. 133 will be noticed.

TABLE XI.

TESTS OF 500-KILOWATT PARSONS TURBO-ALTERNATOR AT CAMBRIDGE.

Trial number.		Effective electrical output in kilowatts.	Consumption of steam.	
			lbs. per hour.	lbs. per kw.-hour.
Trials of Jan. 9	1	518	12,970	25·0
	2	586	14,320	24·4
	3	273½	7,730	28·3
	4	160½	5,320	33·1
	5	0	1,850	—
Preliminary trials of Jan. 8	A.	535	13,350	25·0
	B.	300	8,270	27·6

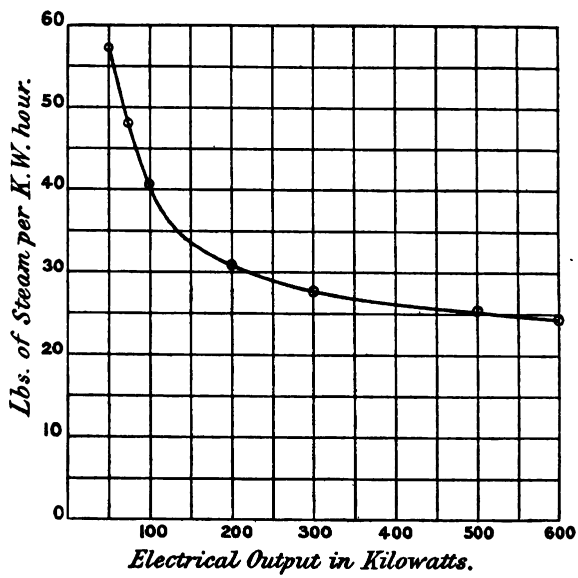


FIG. 132.—Steam Consumption of 500-Kilowatt Parsons Turbo-alternator at Cambridge.

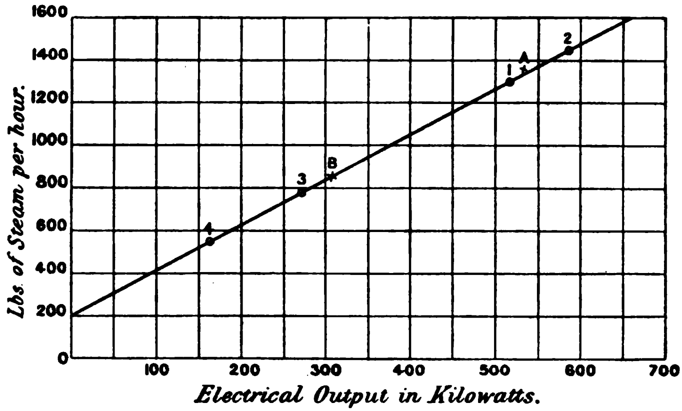


FIG. 133.—Steam Consumption of 500-Kilowatt Parsons Turbo-alternator at Cambridge.

Table XII. supplies particulars of the pressures, temperatures, speeds, etc.

TABLE XII.

TESTS OF 500-KILOWATT PARSONS TURBO-ALTERNATOR AT CAMBRIDGE.

Number of trial.	1.	2.	3.	4.	5.	A.	B.
Electrical output in kws.	518	586	273½	160½	—	535	300
Volts at terminals of generator	2,100	2,150	2,250	2,290	2,280	2,120	2,110
Speed in revolutions per minute	2,670	2,740	2,630	2,590	2,580	2,880	2,800
Air-pump discharge, lbs. per hour	12,970	14,320	7,730	5,320	1,850	13,350	8,270
Air-pump discharge, lbs. per kws. per hour	25·0	24·4	28·3	33·1	—	25·0	27·6
Pressure at stop-valve, lbs. per sq. in.	148	145	151	151	121	145	150
Vacuum in condenser, inches	27·8	27·9	28·2	28·3	28·3	26·6	27·6
Vacuum in turbine cylinder, inches	25·7	25·4	27·2	27·8	28·1	25·1	26·2
Temperature of air-pump discharge, ° F.	74	76	57·5	56	54	90	68
Temperature of circulating water, inlet, ° F.	40	40	38	39	36	41	39
Temperature of circulating water, outlet, ° F.	71	72·5	60	57	46	91	71
Barometer, inches ...	29·93					29·99	

In January, 1900, tests were made at the works of Messrs. C. A. Parsons and Co., Newcastle-on-Tyne, of a 1000-kilowatt

turbine-generator, constructed by that firm for the electric station of the city of Elberfeld. This machine is shown in Plate I. The tests were conducted by W. H. Lindley, Esq., M.Inst.C.E., and Professors Schröter and Weber of the Polytechnicum, Zurich. Steam was supplied by one Babcock and Wilcox boiler, two marine boilers, and a locomotive boiler. A Babcock and Wilcox superheater with independent firing was introduced into the main steam-pipe. The machine was loaded with a water resistance consisting of four electrodes immersed in four iron vessels fitted with water coolers, while an auxiliary adjustable water resistance was employed to regulate the load.

The tests extended over three days, exclusive of a preliminary trial, and the results as regards steam consumption are given in Table XIII.

TABLE XIII.

TESTS OF 1000-KILOWATT PARSONS STEAM TURBO-ALTERNATOR FOR ELBERFELD CORPORATION.

Number of series.	Amount of load.	Exact value in output in kws.	Steam consumption per kw.-hour.		Steam consumption in one hour.
			lbs.	kgs.	kgs.
A.	Preliminary trial	1172·7	18·22	8·26	9,689
II.	Overload	1190·1	19·43	8·81	10,485
I.	Normal load	994·8	20·15	9·14	9,092
III.	Three-quarter load	745·3	22·31	10·12	7,542
IV.	Half load	498·7	25·20	11·42	5,695
V.	Quarter load	246·5	33·76	15·31	3,774
VI.	No load with alternator excited	0	—	—	1,844
VII.	No load without excitation ...	0	—	—	1,183

The same steam-pressure and the same amount of superheat were not used in all the trials. The steam consumption was, therefore, calculated by the experts conducting the tests for a steam temperature of 197·3° C., this being a superheat of

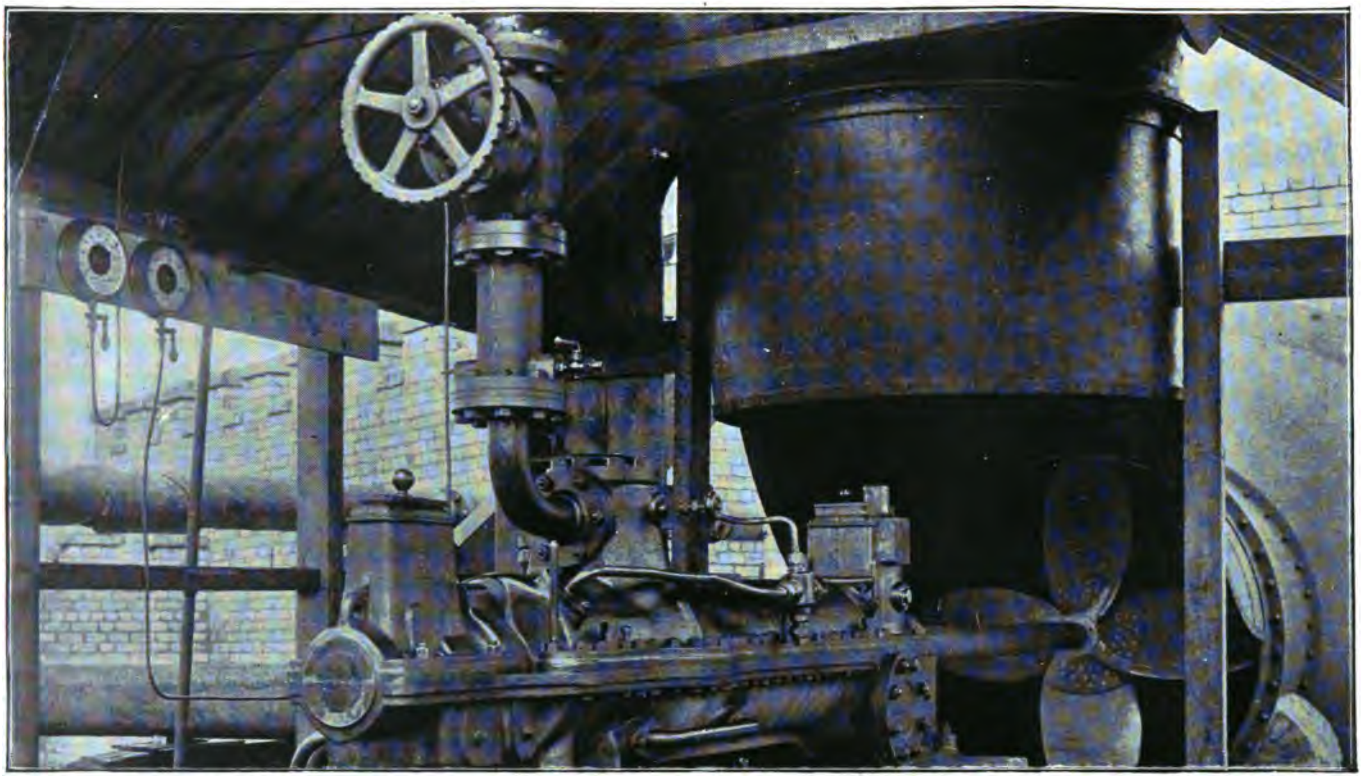


Plate VII.—Ventilating Fan driven by a Parsons Steam Turbine at Clara Pit, Durham.

14·3° C.; and, to enable a comparison to be made with the steam consumptions of engines working with saturated steam, the equivalent consumptions for saturated steam at eleven atmospheres were calculated. The results are given in Table XIV.

Fig. 134 shows the steam consumption graphically.

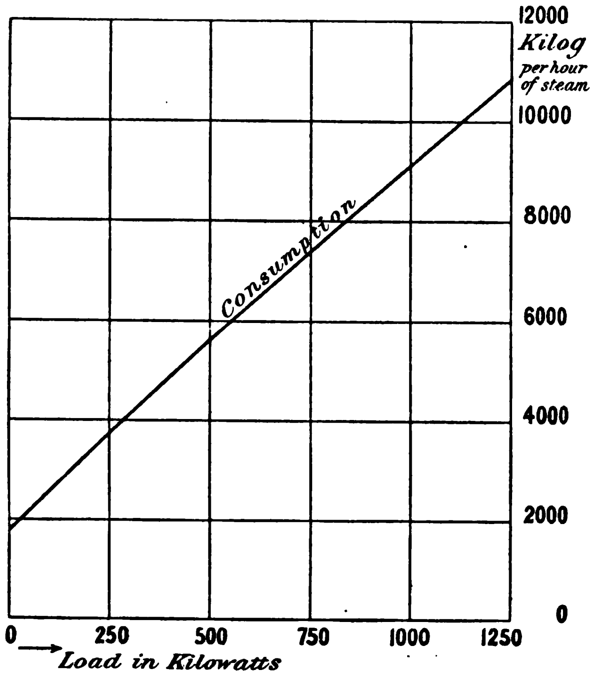


FIG. 134.—1000-Kilowatt Parsons Turbo-alternator. Diagram of total steam consumption per hour.

Table XV. shows the variation in the speed between no load and full load. The number of revolutions per minute was obtained by noting the time occupied by 200 revolutions of the driving-wheel of the valve-gear and air-pump, this driving-wheel rotating at one-eighth of the speed of the turbine.

TABLE XIV.

Tests of 1000-Kilowatt Parsons Turbo-alternator.

Number of series.	Load in kilowatts.	Average observed steam pressure in $\frac{\text{kgs.}}{\text{cm.}^2}$ absolute.	Corresponding temperature of saturated steam.	Average observed temperature of superheated steam at inlet valve.	Superheating (Col. 5 − Col. 4).	Observed steam consumption per kw.-hour.	Total heat contained in 1 kg. of steam at observed steam pressure.		Measured consumption of heat per kw.-hour (Col. 9 × Col. 7).	Corresponding consumption of saturated steam per kw.-hour (Col. 10 ÷ Col. 8).	Corresponding consumption of steam at 11 kgs. absolute and at 14·3° superheating (1 kg. of steam = 669·2 cal.).	Corresponding consumption of saturated steam at 11 atms. absolute (1 kg. of steam = 662·3 cal.).
							In saturated condition.	In superheated condition.				
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
	kilowatts.	$\frac{\text{kgs.}}{\text{cm.}^2}$	° C.	° C.	° C.	kgs.	calories.	calories.	calories.	kgs.	kgs.	kgs.
II.	1190·1	10·11	179·3	189·5	10·2	8·81	661·1	666·0	5,867	8·87	8·76	8·86
I.	994·8	10·47	180·9	192·0	11·1	9·14	661·7	667·0	6,096	9·21	9·11	9·20
III.	745·3	10·76	182·0	190·0	8·0	10·12	662·0	665·8	6,738	10·18	10·07	10·17
IV.	498·7	10·40	180·6	209·7	29·1	11·42	661·6	675·6	7,715	11·66	11·53	11·66
V.	246·5	10·14	179·4	196·4	17·0	15·31	661·2	669·4	10,248	15·50	15·31	15·47
VI.	No load with excitation.	10·34	180·3	193·0	13·3	per hour. 1844	661·5	667·8	per hour. 1,231,423	per hour. 1861	per hour. 1840	per hour. 1859
VII.	No load without excitation.	10·49	181·0	194·5	13·5	1183	661·7	668·2	790,481	1194	1181	1194

TABLE XV.

1000-KILOWATT PARSONS TURBO-ALTERNATOR. VARIATION IN SPEED BETWEEN NO LOAD AND FULL LOAD.

Time.	Load.	Steam pressure.	Vacuum in condenser.	Potential of alternator.	No. of revolutions as counted. No load.	No. of revolutions as counted. Full load.	Variation in the number of revolutions.	Variation per cent.
h. m.	kws.	lbs.	mm.	volts.				
10 44-45	0	150	—	3705	1482	—	—	—
11 16-17	1020	140	693	3960	—	(1433)	(−49)	(3·3)
0 19-20	1035	140	691	3950	—	1424	−58	3·9
11 28-30	0	150	712	3900	1486	—	+62	4·3
11 35-36	1040	145	696	4060	—	1429	−57	3·8
11 44	0	140	712	3880	1472	—	+43	3·0
0 48	960	140	698	4045	—	(1433)	(−39)	(2·6)
0 52	1058	140	693	4040	—	1429	−43	2·9
—	—	—	—	Average	1480	1427	53	3·6

Fig. 135 shows the effect on the speed of governing with a centrifugal governor with an increasing load, while Fig. 136

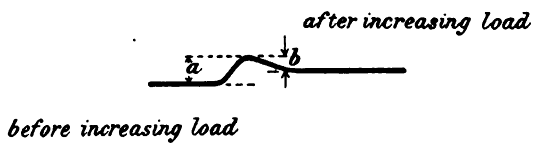


FIG. 135.—Increasing Load.

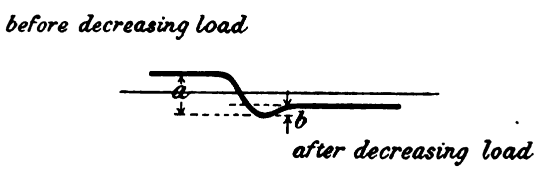


FIG. 136.—Decreasing Load.
Variation in speed with centrifugal governor.

shows the same with a decreasing load. Table XVI. gives a summary of the results, the numbers in the fifth, sixth, and seventh columns referring to the distances marked on the diagrams (Figs. 135 and 136).

TABLE XVI.

Test.	Average of all values of load.	Limits: Variation in the load			Variation in speed.			Variation in potential.			Average of variations in the load. Kilowatts.
		within the limit of	in per cent.		Average.			In per cent.		Average.	
	kilowatts	kilowatts.	max.	min.	a	b	a–b	+	–	volts.	From ⇄ To
IXa	957	1086–840	19·5	16·3	1·75	0·67	1·08	1·29	1·20	50	1050 ⇄ 864
IXb	694	790–590	26·7	16·4	1·28	0·65	0·63	1·19	1·35	51	766 ⇄ 623
IXc	497	590–400	47·5	30·5	1·36	0·73	0·63	1·32	1·28	53	590 ⇄ 404
IXd	405	500–306	63·4	36·0	1·62	0·86	0·75	1·35	1·41	56	490 ⇄ 312
IXe	251	292–204	43·1	26·9	1·37	0·63	0·74	1·34	1·29	51	292 ⇄ 210 and back.

As the average potential may be taken at 4000 volts, the actual variation of 52 volts on the average amounts to **1·3** per cent. of the initial potential.

TABLE XVII.

Test.	Average of all values of load.	Limits: Variation in the load			Speed.					Variation in voltage.			Average of variations in the load. Kilowatts.	
		within the limits of	in per cent.		Variations.			Average. (a–b)		In per cent.		Volts.		
	kilowatts.	kilowatts.	max.	min.	a	b	d	+	−	+	−		From	To
Xa	281	336–222	51·3	27·5	—	—	—	0·158	0·227	1·05	1·10	43	230 ⇄	332
Xb	492	616-380	62·1	34·4	0·31	0·22	1·32	0·84	0·75	1·10	1·15	45	382 ⇄	601
Xc	714	(900)–580	55·2	12·2	0·24	0·20	1·29	0·99	0·73	1·11	1·10	44	611 ⇄	818
Xd	900	1016–790	30·6	19·3	0·21	0·27	1·26	0·86	0·80	1·06	1·12	45	797 ⇄	1007
													and back.	

The average variation in the potential amounts to 44 volts, *i.e.* to **1·1** per cent. of the initial voltage.

Figs. 137 and 138 and Table XVII. show the effects of governing with an electrical governor.

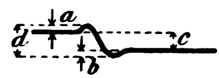


FIG. 137.—Increasing Load.

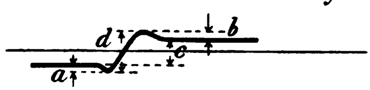


FIG. 138.—Decreasing Load.
Variation in speed with electrical governor.

It will be noticed that the centrifugal governor increases the speed with diminishing load and reduces the speed with increasing load, while the action of the electrical governor is the reverse.

CHAPTER XII.

THE STEAM TURBINE APPLIED TO THE PROPULSION OF VESSELS.

THE success of the Parsons steam turbine on land led to the formation of a company in the beginning of 1894 for applying the steam turbine to marine purposes. This pioneer syndicate—the Marine Steam Turbine Co.—at once commenced experimental work, and the *Turbinia* was produced. It had often previously been proposed to use a steam turbine for the propulsion of vessels at sea; but, as far as the author is aware, no steam turbine was ever before fitted on board a vessel for this purpose. The same difficulty now arose with the marine steam turbine as had arisen with turbines previously made for use on land—namely, of running the turbine economically at a sufficiently low speed. In the driving of alternators a high speed is usually an advantage, except when it becomes so excessive as to occasion dangerous stresses due to centrifugal force. With screw propellers, however, the case is very different. The existence of cavitation with high velocities of screw propellers was not unknown at the time the *Turbinia* was built; but the importance of it with propeller-blade velocities such as those tried in the *Turbinia* was not appreciated. The trials of the *Turbinia*, however, clearly demonstrated that an ordinary propeller could not be run with any degree of efficiency above a certain velocity. Beyond this limiting

velocity (the exact value of which depends on the size and form of the propeller) an almost perfect cylindrical vacuum is formed around the propeller, causing great loss of power.

As a steam turbine could not be run economically except at a high velocity—above the limiting velocity of a propeller—the difficulty arose of getting an efficient combination. With a low velocity the steam consumption was excessive; with a high velocity the waste of power by the propeller was enormous.

The designers of the *Turbinia* and her propelling gear, however, energetically and scientifically grappled with the difficulty. Trials were made with screws of various patterns, a spring torsional dynanometer was constructed and fitted between the turbine and the propeller-shaft to measure the actual torque, and a series of experiments were carried out in a tank with model propellers, which were illuminated by the light from an arc lamp thrown on to them for a single instant in each revolution. At length, after a great amount of labour, the efforts of the experimenters were crowned with success, a combination and arrangement of turbines and screw propellers being obtained which gave excellent results—results as good as the most optimistic of well-wishers had ever hoped for.

The solution of the difficulty was found in dividing up the power into three turbines driving three propeller-shafts. Each shaft carried three propellers of a special form. As the economic speed of a turbine depends on the difference of pressure of the entering and exhausting steam, it will be obvious that, by dividing the total range of pressure into three parts—that is, in expanding the steam only about one-third in each turbine—the minimum economic speed of each machine could be very much reduced—in fact, reduced to about one-half. The propeller-shafts could thus rotate at one-half the speed. In

Plate VIII.—The Pioneer of Marine Steam Turbine Propulsion.

addition to this, the employment of so large a number of propellers—nine in all—allowed each to be of small size, and therefore allowed the tips of the blades to revolve in circles of small diameters. By thus reducing both the size and the angular velocity of the propellers, and giving them a suitable design, their efficiency was brought quite up to the normal. The result was that the *Turbinia* attained a speed—33 to 34 knots—never before reached by any vessel.

The length of the *Turbinia* is 100 feet and the beam 9 feet. The displacement is 44½ tons, which if made up as follows:—

Main engines, 3 tons 13 cwts.	
Total weight of machinery and boiler, screws and shafting, tanks, etc.	22 tons
Weight of hull complete	15 „
Coal and water	7½ „
Total displacement	44½ „

Steam is supplied by a water-tube boiler, and enters the first turbine cylinder at a pressure of 170 lbs. per square inch. The heating surface of the boiler is 1100 square feet, and the grate area 42 square feet. The stoke-holds are closed, and draught is furnished by a fan coupled directly to the engine-shaft. 4200 square feet of cooling surface are provided in the condenser. The fresh-water tank and hot well contain about 250 gallons of water. The auxiliary machinery consists of main air-pump and spare air-pump, auxiliary circulating pump, main and spare feed-pumps, main and spare oil-pumps, and bilge ejectors.

The engine cylinders lie close to the bottom of the boat, and are bolted directly to small seatings on the frames. The reaction of the propellers and the axial thrust of the steam on the

rotating parts of the turbine are arranged as far as possible to balance one another; but small-thrust bearings are provided in the turbine bearings to withstand any difference or error of balance. Lignum-vitæ bearings are used for the propeller-shafts. Astern motion is given to the vessel by means of a reversing turbine situated on the central shaft.

The hull of the boat is built of mild steel plates, varying in thickness from $\frac{3}{16}$ inch at the bottom to $\frac{1}{16}$ inch at the sides near the stern. Water-tight bulkheads divide the vessel into five compartments.

The success of the *Turbinia*, which was only built for experimental and demonstrative purposes, led to the formation under the same directorate of a larger company—the Parsons Marine Steam Turbine Co., Ltd.—and the construction of the ill-fated torpedo-boat destroyers, *Viper* and *Cobra*. Of these the first was built to the order of the British Admiralty, who subsequently purchased the other after completion.

The *Viper* was 210 feet long, 21 feet beam, and 12 feet 9 inches moulded depth, the hull being constructed with the standard Admiralty scantlings for 30-knot destroyers, and further strengthened in parts for the higher speeds contemplated. The displacement was 350 tons. There were four shafts and two propellers on each shaft, the after propeller on each shaft having a slightly greater pitch than the forward one. On each side of the vessel a high-pressure turbine drove the outer and a low-pressure turbine the inner shaft. The inner shaft on each side was also fitted with a reversing turbine, the two reversing turbines being capable of driving the vessel astern at a speed of 15 knots. Plate IX., reproduced by kind permission from *Engineering*, shows one set of turbines. The cylinder on the left is the high-pressure turbine, and the one

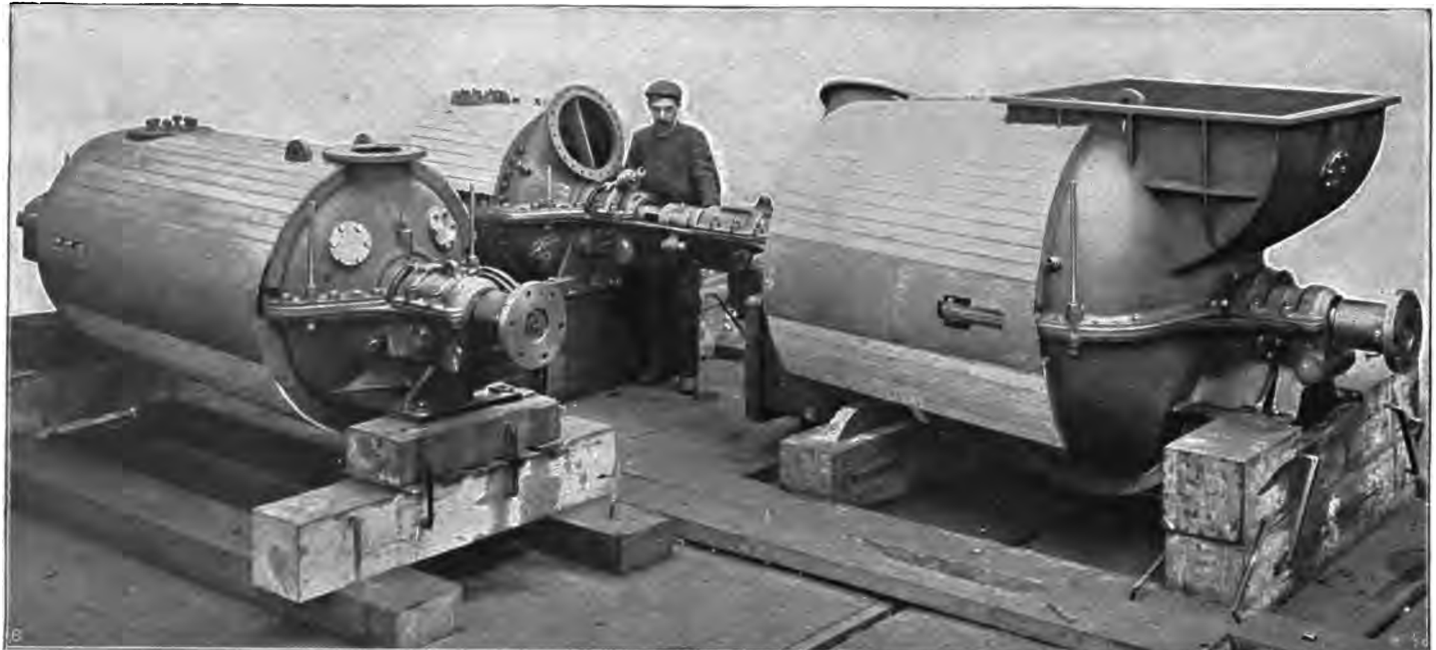


PLATE IX.—ONE SET OF ENGINES FOR H.M. TORPEDO-BOAT DESTROYER "VIPER" SUPPLIED BY THE PARSONS MARINE STEAM TURBINE COMPANY, LIMITED.

From "Engineering," by kind permission.

to the right on the other shaft is the low-pressure turbine, which receives the steam which exhausts from the high-pressure cylinder. The small cylinder at the back is the reversing turbine. The set of engines for the other side of the vessel was similar. Steam was supplied by four Yarrow boilers, having a total heating surface of 15,000 square feet, and a total grate area of 275¾ square feet. The thrust of the propellers was arranged to balance the thrust of the turbines. The fittings were constructed to satisfy Admiralty requirements, and were much the same as those of other destroyers. The diameter of each high-pressure cylinder was 35 inches, and of each low-pressure cylinder 50 inches. The weights of boilers and machinery are as follows :—

Boiler-room weights with water in boilers ...	120 tons
Engine-room weights with auxiliary gear and water in condensers	65 „
Propellers, shaftings, etc.	8 „
Total	193 „

Although the contract for the whole vessel was given by the Admiralty to the Parsons Marine Steam Turbine Co., Ltd., that firm, while themselves making and fitting on board the engines, sublet the contract for the hull and boilers to Messrs. Hawthorne, Leslie and Co.

On her official steam trials under the direction of the Admiralty officials, the *Viper* easily attained a speed of 33·838 knots on a three-hours' run. At this speed, the consumption of coal was 11 tons 9 cwt. 1 qr. 9 lbs., or 25,685 lbs. per hour. On a three-hours' trial at 31·118 knots, the coal burned per hour was 19,846 lbs.

At a preliminary trial instituted by her contractors, the *Viper*, with a displacement of 380 tons, attained a mean speed on two runs with and against the tide of 36·849 knots. The mean speed for an hour's run alternately with and against the tide was 36·581 knots, the mean revolutions being 1180 per minute. The steam pressure during the six-hours' run ran up to 200 lbs., and the mean air-pressure in the stoke-holds was 4½ inches. The speed was changed from 14 knots to 36·585 knots in twenty minutes.

The *Viper* was wrecked, it will be remembered, off Alderney in a fog, during the naval manœuvres in the summer of 1901.

The *Cobra* was built by Sir W. G. Armstrong, Whitworth and Co., Ltd., and engined by the Parsons Marine Steam Turbine Co., Ltd. This boat was slightly larger than the *Viper*, although of less beam (the small beam being noticeable in many war-vessels of Elswick design). The length was 223 feet 6 inches; beam, 20 feet 6 inches; draught, 6 feet; displacement, 400 tons. The *Cobra* foundered during a gale on September 18, 1901, while being taken from the Tyne to Portsmouth Dockyard to undergo trials by the Admiralty. She was not quite so fast a vessel as the *Viper*.

The first merchant steamer to be propelled by steam turbines is the *King Edward*, which commenced running in July, 1901. This vessel was built by Messrs. William Denny and Bros., of Dumbarton, and is engined with Parsons' turbines.

The dimensions of the vessel are as follows: length, 250 feet; beam, 30 feet; moulded depth, 10 feet 6 inches to the main deck, and 17 feet 9 inches to the promenade deck. Steam is supplied by a double-ended return-tube Scotch boiler of the usual marine type, having four furnaces at each end. There are three propeller-shafts, of which the two outer ones each

carry two propellers, the central shaft being provided with only one. A high-pressure turbine is situated on the central shaft, in which turbine the steam supplied at 150 lbs. is expanded about 5-fold, and then passes to two low-pressure turbines on the wing shafts, where it is expanded about 25-fold, the total expansion, therefore, being about 125-fold. Reversing is done by two turbines situated in the exhaust ends of the casings of the main low-pressure turbines. Steam can be supplied direct to the low-pressure cylinders, and the high-pressure turbine and its shaft cut out of use in order to obtain greater manœuvring power for negotiating piers. The weight of the motors, condensers with water in them, steam-pipes, auxiliaries connected with the propelling machinery, shafting, propellers, etc., is 66 tons, which is very much less for the power developed than the propelling machinery of reciprocating-engine, paddle-propelled passenger steamers of the same type.

The *King Edward* was employed for passenger traffic between Fairlie and Campbeltown in the summer of 1901, and gave great satisfaction. The turbines produce no vibration whatever, a slight vibration aft being due to the propellers.

In the trials of the *King Edward*, on June 26, 1901, on the Clyde, a mean speed of 20·48 knots was obtained on several runs over the measured mile at Skelmorlie. The mean revolutions at this trial were 740 per minute. The steam-pressure at the boilers was 150 lbs., and the vacuum 26½ inches. The air-pressure in the stoke-hold was equal to 1½ inches of water.

Figs. 139–142 illustrate a propeller-shaft support, recently patented by Messrs. Parsons and Wass, as applied to a vessel with a flat bottom upwardly inclined at the stern. Fig. 139 shows the support in end elevation, partly in section. Fig. 140 is a side elevation of part of the vessel with the support and

propeller-shafts. Fig. 141 is a section on a line below the part of the vessel shown in Fig. 140. The support consists of two Y-shaped brackets of elliptical section, as shown at *a*, Fig. 141. The approaching arms of the two brackets are connected by a boss, while each of the outside arms also carries a boss. These bosses are lined with lignum-vitæ or white metal. Each bracket carries a sole, B, which is placed in a socket, C, in a sole-plate, D, which is machined to receive it. The sole-plates are preferably formed of cast steel, and are permanently attached to

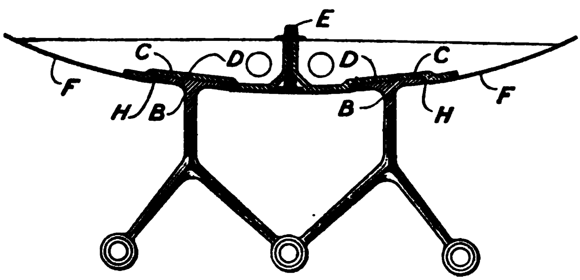


FIG. 139.—Propeller-shaft Support of Parsons and Wass: Sectional end elevation.

the framing E and plates F of the vessel, the plates being cut away to allow of the insertion of the soles. If the brackets are formed of aluminium bronze, manganese bronze, or gun-metal, strips H are provided round the soles to prevent corrosion. The end support for the central shaft is shown in Fig. 142. An arrangement of brackets for four propeller-shafts is shown in Fig. 143.

It will be seen that these propeller-shaft supports will offer very little resistance to passage through the water, and will be light and easily fitted correctly to the vessel.

Mr. Parsons states that he has found that the cavitation which attends high-speed propellers occurs principally in two places, namely, at the back faces of the blades near the tips, and

around the conical tip of the propeller-boss behind the blades. To obviate or lessen cavitation at the blade-tips, Mr. Parsons

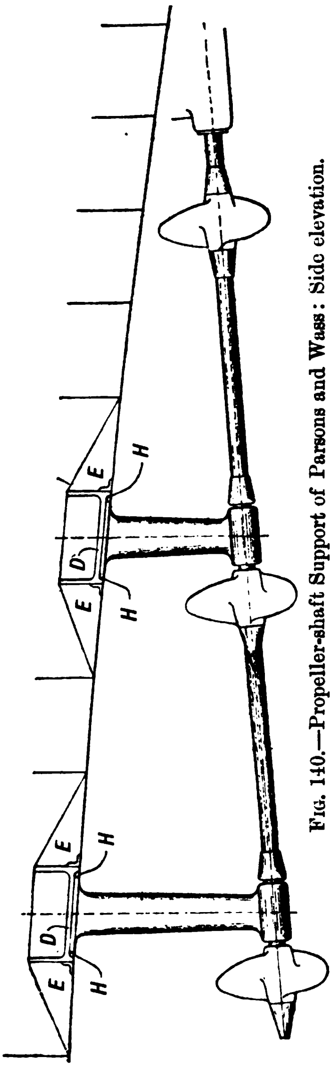


FIG. 140.—Propeller-shaft Support of Parsons and Wass: Side elevation.

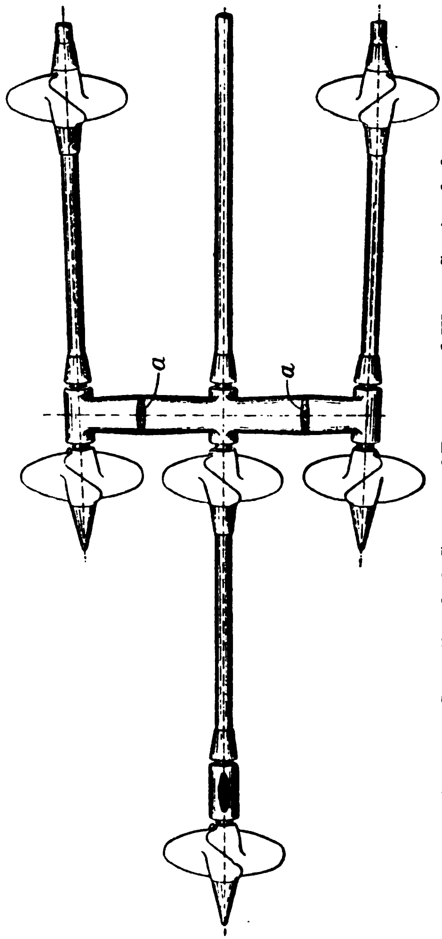


FIG. 141.—Propeller-shaft Support of Parsons and Wass: Sectional plan.

prefers to form the blades with diminishing pitch near the tips.

A device for diminishing cavitation round the conical end of the boss has been patented by Mr. Parsons, and is shown applied to a propeller in Fig. 144. Vanes *v* are fixed on the

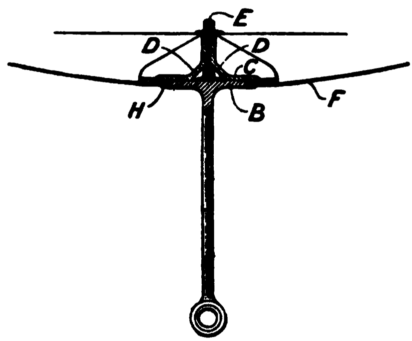


FIG. 142.—Propeller-shaft Support of Parsons and Wass: Rear support of centre shaft.

conical end *x*, the vanes being parallel, or nearly so, to the axis of the shaft *y*. Fig. 145 is a cross-section through the cone and vanes. The water put into rotation by the propeller-blades

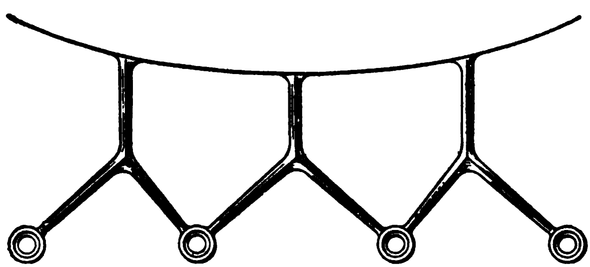


FIG. 143.—Support for Four Propeller Shafts.

closes in on the cone *x*, but tends to retain its velocity. It therefore rotates with a greater angular velocity than the cone. The vanes *v* are, therefore, considered to produce two beneficial results. Firstly, some of the kinetic energy of the rotating water is given up to the shaft which it helps to rotate; and

secondly, owing to the diminution of the velocity of the water rotating round the shaft, centrifugal force is reduced, and the

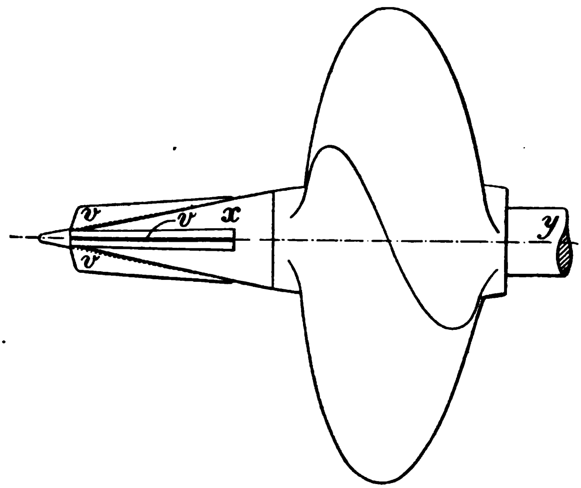


FIG. 144.—Parsons' Construction of Propeller Boss to diminish Cavitation.

water closes in more readily, and, pressing on the cone x, imparts an additional forward thrust to the shaft.

The steam turbine possesses several advantages over the

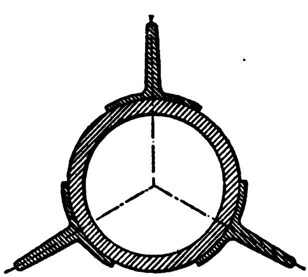


FIG. 145.—Cross-section of Boss.

reciprocating engine for marine propulsion. In the first place, there is the absence of vibration—an important point, both as regards comfort in passenger steamers and as regards accuracy

of gun-fire in naval vessels. Then there is a distinct saving in weight. This is not so marked in vessels of the destroyer type, where the engine-room weights are cut down to an abnormally small amount, as in larger vessels, and especially in the mercantile marine. This saving in weight can, of course, be used either in increasing the engine power, and consequently the speed, of the vessel, or in adding to its carrying capacity. The low situation of the engine-room weights in a turbine-propelled vessel also tends to improve the stability, and, in the case of a war-vessel, places the engines in a more protected position.

APPENDIX

BRITISH PATENTS FOR OR RELATING TO STEAM TURBINES FROM THE EARLIEST RECORDS UP TO THE END OF 1899.

When inventions have been communicated from abroad, the names of the communicators are printed within parentheses.

1784.

1,426 . . Kempelen.
1,432 . . Watt.

1791.

1,812 . . Sadler.

1805.

2,887 . . Miller.

1809.

3,289 . . Noble.

1815.

3,922 . . Trevithick.

1823.

4,793 . . Peel.

1830.

5,910 . . Grisenthwaite.
5,961 . . Ericsson.

1831.

6,120 . . Hobday.

1834.

6,720 . . Craig.

1836.

7,242 . . Perkins.

1837.

7,305 . . Elkington.

1838.

7,554 . . Heath.
7,797 . . Burstall.
7,854 . . James.

1840.

8,474 . . Williams.
8,572 . . Cordes and Locke.

1841.

9,116 . . Jones.

1842.

9,354 . . Pilbrow.

1843.

9,658 . . Pilbrow.
9,902 . . Walther.

1844.

10,189 . . McIntosh.

1845.

10,765 . . Meade.

1846.

11,044 . . Taylor.
11,352 . . Bessemer.

1847.

11,800 . . Von Rathen.

1848.

12,060 . . Wilson.
12,080 . . Exall.
12,217 . . Stenson.

1850.

13,245 . . Barclay.
13,281 . . Fernihough.

1851.

13,598 . . Andrews.

1852.

14,351 . . Gorman.
149 . . Wheel.
776 . . Presson.
1,083 . . Slate.

1853.

480 . . Nicholls.
735 . . Brown.
2,768 . . Sochet.

1854.

315 . . Tourney.
944 . . Danchell.
1,706 . . Tetley.

1855.

2,747 . . Poulson.

1857.

2,076 . . Ivory.
2,598 . . Lombard.
3,061 . . Parker.

1858.

144 . . J. and E. Harthan.

1859.

805 . . Ivory.
1,041 . . Taylor.

1860.

119 . . Rutchet, Vonwiller, and Seiler.
1,155 . . Boyman.
2,317 . . Budden (Pilkington).

1861.

770 . . Chevillard.
2,457 . . Coffey.
2,953 . . Macintosh.

1862.

552 . . Parker.
1,568 . . Brakel, Hoehl, and Gunther.
3,252 . . Braddock.
3,283 . . Budden (Pilkington).

1863.

1,160 . . Thomson.
2,355 . . Lloyd.
2,692 . . Verran.

1864.

502 . . Southam.
2,596 . . Newton.
2,779 . . Galloway.

1865.

949 . . Brookes (Perrigault, Farcot, Farcot, Farcot, Château, and Farcot).
2,130 . . Stevenson (Venzano).

1866.

891 . . Wenner.
1,206 . . Newton (Farcot and Perrigault).
1,822 . . Fraser.
2,270 . . White (Sellier and Hermant).
3,289 . . Newton (Harris).

1867.

646 . . Clark, W. (Lemley, G. W.).
984 . . Moll, J. A.

1868.

784 . . Parker, J.
883 . . Beech, T. S. L.
1,732 . . Newton, W. E. (Boorman, J. M.).
2,320 . . Brooman, C. E. (Hielakker, J. V.).
2,680 . . Hunter, J. M.
3,146 . . Robertson, J.
3,307 . . Meldrum, R.
3,933 . . Lake, W. R. (de Amezaga, F.).

1869.

68 . . Legg, R.
208 . . Cook and Watson.
1,159 . . Brooman, C. E. (Goguel, E. F. A.).
1,748 . . Clark, A. M. (Lesnard, F.).
2,476 . . Mayall, J. J. E.
2,648 . . Muller, J. A.
2,830 . . Walker, W., and Davies, D.
3,267 . . Gorman, W.
3,642 . . Outram, J.
3,705 . . Bourne, J.

1870.

1,537 . . Astrop, W.
1,904 . . Lake, W. R. (Smith, J. Y.).
2,086 . . Scott, B. C.

1871.

1,736 . . Griffin, G. F.

1872.

2,188 . . Lake, W. R. (Harris, J.).
3,134 . . Robertson, J.
3,835 . . Cotter, R. H.

1873.

1,493 . . Burnett, W.
3,161 . . Baldwin, T.

1874.

706 . . Teulon, A.
3,961 . . Louche, J. H.

1875.

51 . . Turnock, J.
67 . . Boyman, R. B.
1,676 . . Newton, H. E. (Babbitt, B. T.).
1,848 . . Clark, A. M. (de Romilly, H. F. L. W.).
2,184 . . Preiswerk, L.
4,324 . . Preiswerk, L.

1876.

1,224 . . Pope, A.
1,549 . . Cotton, Sir A.
2,368 . . Clark, A. M. (Dufort, J. H.).
3,483 . . Apperly, J.
3,841 . . Harris, J.

1877.

862 . . Apperly, J.
2,434 . . Lake, W. R. (Averseng, M. A. T.).
2,864 . . Smith, T. J. (Penning, G. A. de).

1878.

1,985 . . Brydges, E. A. (Bazin, R.).
4,293 . . Apperly, J.
4,596 . . Lumley, H. R.
4,628 . . Mills, B. J. B. (Gfeller, J.).
4,682 . . Tuckey, T.

1879.

409 . . Abel, C. D. (Binzer, J. von, and Bentzen, E.).
2,673 . . Davies, P.
3,521 . . Rigg, A.
5,022 . . Cutler, W. H.

1880.

17 . . Jensen, P. (Hahn, E. J.).
1,222 . . Prowett, W.
2,496 . . Howson, J. T., and Tate, W.
2,609 . . Nedden, F. zur.
3,522 . . Temple, G.
3,980 . . Jensen, P. (Hahn, E. J.).
4,160 . . Lake, W. R. (Cole, J. W.).

1881.

177 . . Imray, J.
255 . . Willet, T.
369 . . Temple, G.
981 . . Willet, T.
2,857 . . Leverkus, K. W. A.
5,237 . . Newton, H. E. (Desruelles, L. A. W., and Carlier, C. F.).

1882.

2,166 . . Charlton, G., and Wright, J.

1883.

911 . . Capell, G. M.
1,655 . . Engel, F. H. F. (Laval, G. de).
4,245 . . Johnson, J. H. (Delaurier, E. J.).
5,233 . . Lake, W. R. (Emmanuel, C.).

1884.

5,610 . . De Laval, G.
6,734 . . Parsons, Hon. C. A.
6,735 . . Parsons, Hon. C. A.
12,950 . . Dumoulin, A. J. A.

1885.

1,174 . . Johnson, J. H. (Howell, J. A., and Paine, F. H.).
3,885 . . Last, W. I.
4,483 . . Curtis, N. W.
8,773 . . Howson, J. T.

1886.

1,157 . . Neil, W.
5,647 . . Thévenet, J.
13,805 . . Tongue, J. G. (Brunner, A.).
13,949 . . Whittle, W.
16,020 . . De Laval, G.

1887.

5,312 . . Parsons, C. A.
9,591 . . Gwynne, J. E. A.
12,488 . . McConnell, J.

1888.

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